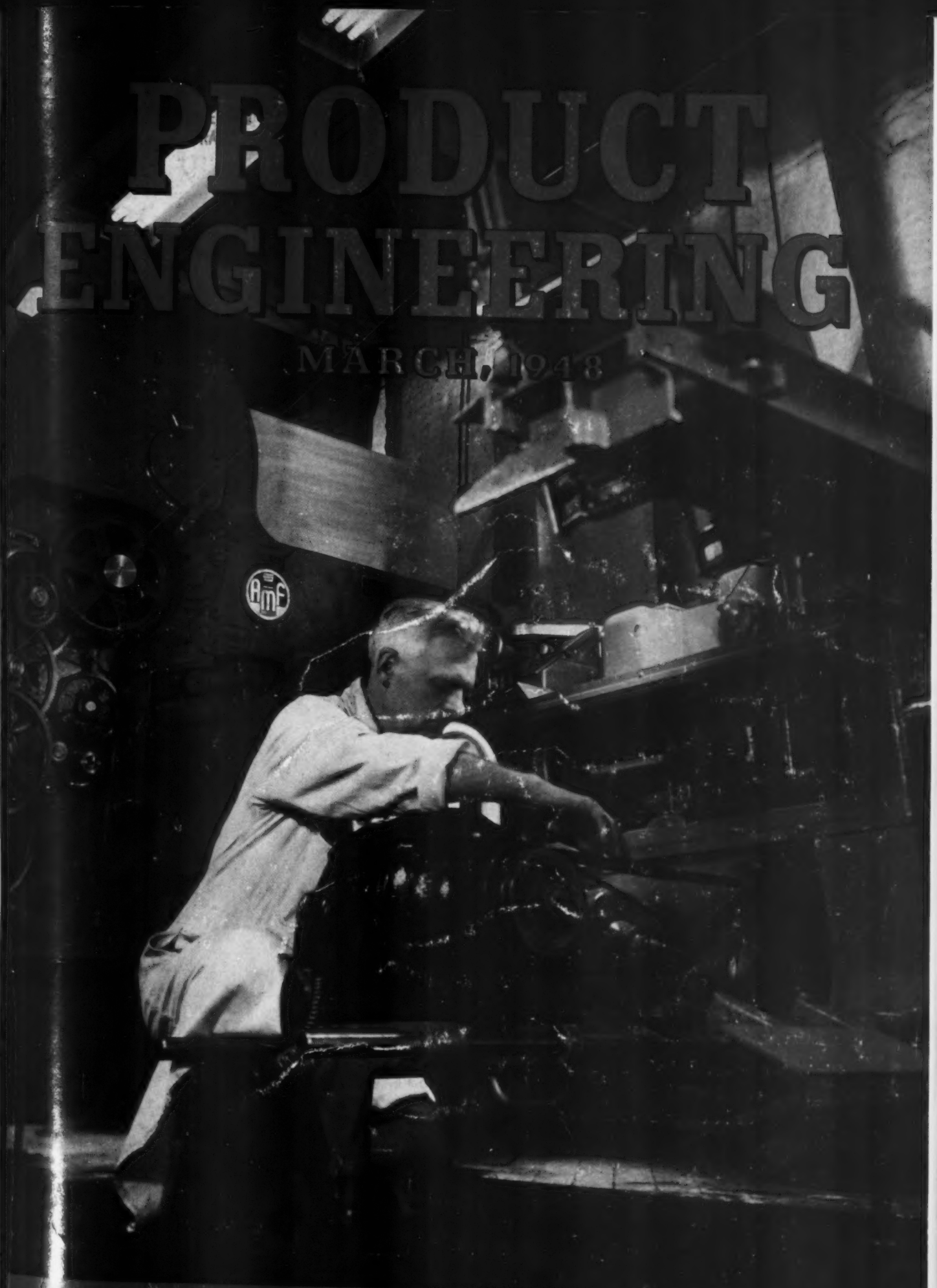
PRODUCT ENGINEERING

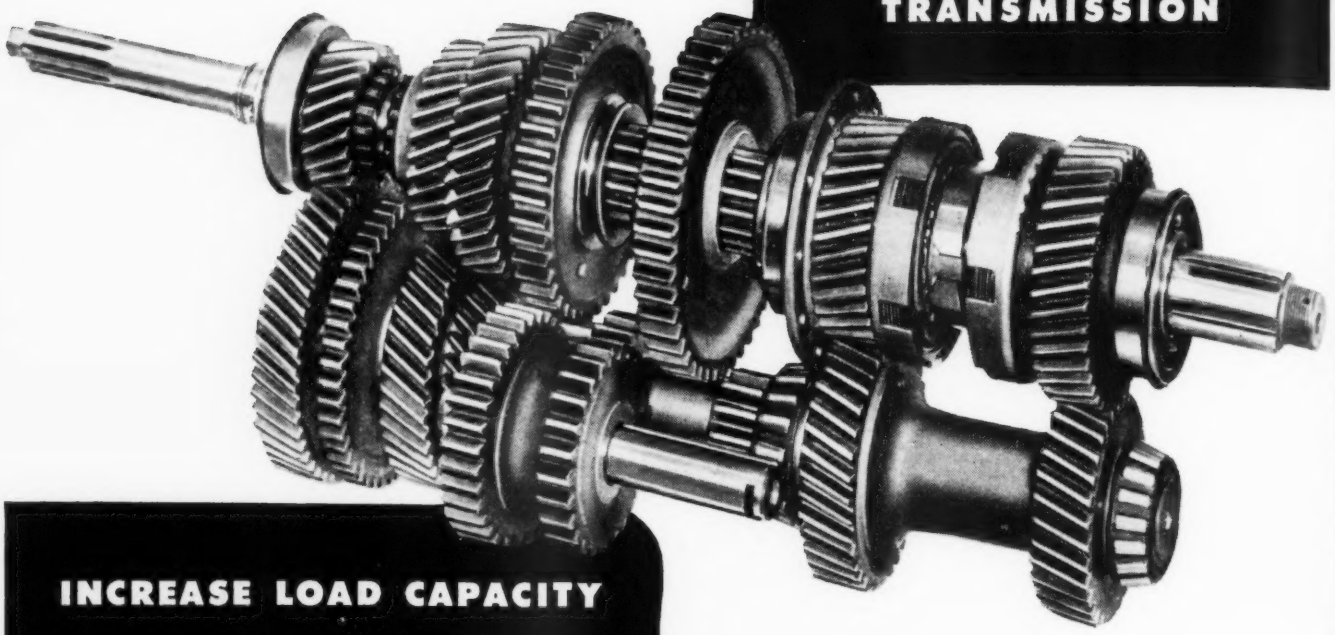
MARCH, 1948



IN THIS ISSUE—Motor Driven Machines + Designing With Dollars + Gear Contact Ratio + Sources of Noise + Bearing Selection + High Temperature Ceramics

Take a Tip on Gears

...from this new *Mack* 10-SPEED TRANSMISSION



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RAISE IMPACT STRENGTH

ADD WEAR-RESISTANCE

...with Nickel Alloy Steels

These gears demonstrate basic advantages secured by using Nickel alloy steels . . .

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Over the years, International Nickel has accumulated a fund of useful information on the selection, fabrication, treatment and performance of engineering alloy steels, stainless steels, cast irons, copper-base and other alloys containing Nickel. This information is yours for the asking. Write for "List A" of available publications.

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Product Engineering

VOLUME 19 NUMBER 3

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JOHN VAN HAMERSVELD. Product cost usually comes down if designers have access to and use reliable cost data. The author discusses the elements which make up product cost and the organization of cost data. Numerous curves, charts and tables are shown which illustrate the ways in which this information is prepared for designers.

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APRIL HIGHLIGHTS

EXPERIMENTAL STRESS ANALYSIS. Lighter, stronger, more economical structures have been developed through experimental techniques using photoelasticity, brittle coatings, electric and mechanical strain gages. In the hands of a capable engineer, these tools often reveal stress distributions and design weaknesses startlingly different from the expected. Although these techniques are used by the aviation and automotive industries as everyday tools, there are many possibilities for profitable applications in other design fields.

"How to Organize for Experimental Stress Analysis" is a 24 page insert that will orient product development engineers in these aids to better design. Advantages, as well as limitations, of the various techniques are discussed. Equipment requirements, costs and applications are given. Design possibilities are illustrated by stress analysis studies on products ranging from firearms through ships.

PRECISION INVESTMENT CASTING, by Edwin Laird Cady. Discussion of the finer points of precision investment casting. The effects of alloys, location of feeders, recommended limits on physical dimensions and the machinability of castings are considered.

METALLIZED CIRCUITS. John T. Collier describes the technique, advantages, limitations and cost of this new process for mass production of electrical circuits.

SIX TYPES OF BRAZING are compared by S. De Domenico and A. Squire, Westinghouse materials engineers. Includes design pointers, influence of materials on brazing method, and physical properties of brazed joints.

SPRING REDESIGN by A. Bodenschatz, I.T.E. Circuit Breaker Co. A time-saving method for redesigning springs of fixed length and spring rate.

CUTTING COST by designing right at the start for minimum scrap in fabricated parts. Down-to-earth guidance from E. F. McCaleb, General Electric Company.

He Created a Masterpiece

With the death of James H. McGraw, Sr., a great personality and publisher passed into history. I will cherish the memories of the intimate contacts I had with him during the few years I knew him. One of our meetings stands out above all others. He and I and a close mutual friend had lunch together. James H. McGraw had just read the book "The Winning of Barbara Worth" and during the luncheon he gave his interpretation of this romance written by Harold Bell Wright. His piercing eyes glittered. There was no hesitation, no repetition, no ambiguity as his resonant voice held our attention.

Immediately after that luncheon, I tried to put into an editorial the ideas that I had just heard. I used as many of his phrases as I could remember.

It is a fitting time to repeat that editorial, which appeared in the November 1938 number of Product Engineering, not because of any merits it may have as a piece of writing, but because it reflects the vision, personality and character of the man who himself created the masterpiece that has put his name among the greatest publishers of all times.

What Makes a MASTERPIECE?

In his book, "The Winning of Barbara Worth," Harold Bell Wright described a great engineering achievement, the transformation of desolate desert wastes into a great fertile valley covered with fields of billowing grain and groves of flowering fruit trees and dotted with cities and hamlets. It is a story packed full of engineering romance and the glories of its accomplishment in creating a masterpiece.

In this story, Harold Bell Wright did not detail the mixture of concrete used, how the forms were built, how the concrete was poured. Nor did he explain how the difficult engineering problems were solved, what formulas were applied, or how an exceptionally intricate engineering difficulty was surmounted. Perhaps Mr. Wright did not possess the engineering knowledge to write about such details. But had such details been included the story would have been spoiled. An achievement is a masterpiece not by virtue of the difficulties encountered in its creation but by virtue of the emotions and inspirations that the accomplishment arouses.

The Cathedrals of Europe, the Last Supper by Leonardo da Vinci, a nation's national anthem, the pyramids of Egypt, the architecture and statuary of Greece—all are masterpieces for what they are and not by reason of the difficulties under which they were created. The same may be said of the steam turbine, the telephone, radio, the machinery of production and construction and the thousands of other creations of engineers.

It is only human that one have a tendency to judge the greatness of his accomplishment by the effort required and the ingenuity exercised. A "tricky" design is always intriguing, especially to the one who developed it. Only too often the engineer permits such thoughts to overshadow the reality of his accomplishment or the lack of it.

Conversely, there is always a tendency to allow the drabness of engineering calculations to blot out its romance and drama. The details of the procedure may obscure the objective. Of course every detail of design must be calculated accurately and considered carefully. The right materials must be used, every part must be correct in size and shape and everything must go together properly. But these are only the tasks, chores perhaps, that are the means to an end—the creation of a masterpiece.

And what is a masterpiece in engineering design? The answer cannot be given in words. If when gazing at the finished product it provokes the same feelings as experienced in the first view of a grand cathedral or an object of fine art—or if it brings forth visions of a great service to humanity, making people happy, giving them greater opportunities, serving their needs or relieving suffering—it is a masterpiece. But if it does not possess any of these attributes, it is just another design, another job out of the way to make room for another one to be tackled.

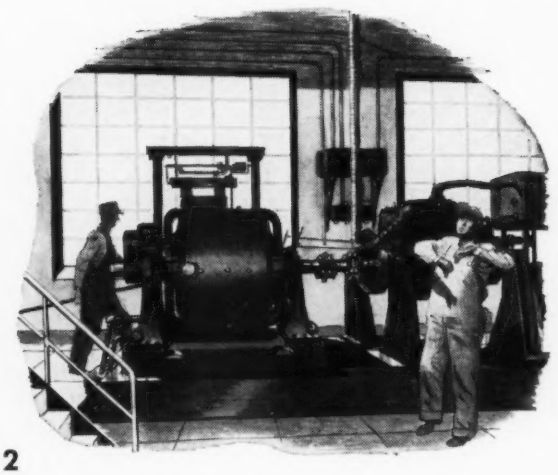
Only those who can visualize and feel the romance and drama in engineering design can create its masterpieces, for every masterpiece embodies the spirit of its creator.

G. F. Jordenholt

Have a case of shock to cure?

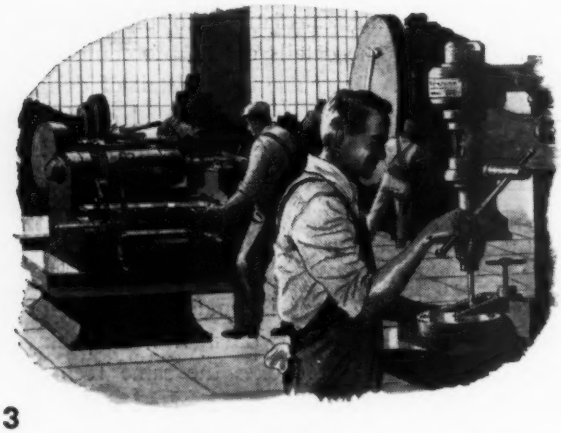
1

"Rev 'er up" is the cry as truck engines roar up to 3000 RPM on giant test stands. It's done to make sure they'll stand the gaff. But occasionally an engine gets ornery — backfires; stops suddenly. Shock! Plenty of it. But what's a backfire to you? Well, what Morse Chain Company did about it may be a boon to *your* business, too . . .



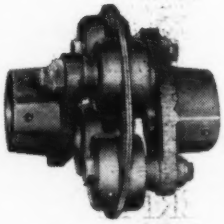
2

Shake, shock and shimmy were the problems Morse Power-Transmission Engineers found on test blocks: a natural for Morse Morflex Coupling. This versatile Morse product absorbs torsional shock, teeth-rattling vibration, and shaft misalignment. Morflex between engines and test blocks reduced bearing wear, protected equipment. This helps you to . . .

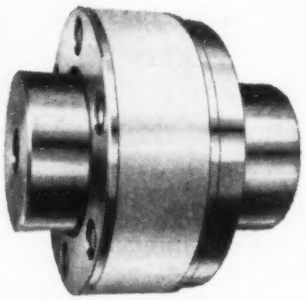


3

Save yourself a headache. Shafts, wheels and machines in your business can be safeguarded. When shaft vibration causes bearing failure it might be prevented by Morflex. *Wherever there is industry* they are adaptable to universal drive connections; allow no power loss; are sealed against dirt and moisture; free of metal-to-metal contact!



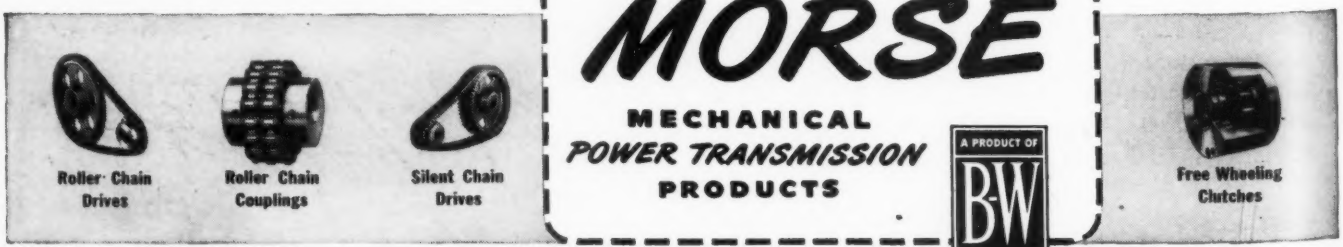
Morflex Coupling



Morflex Radial Coupling

4

Proof of outstanding performance? Call or write Dept. 126, Morse Chain Company, Detroit, Michigan. Hear case histories from experienced Morse Engineers. You'll soon see *how* and *why* Morse Couplings, Roller and Silent Chains, Sprockets and Clutches have been profitably applied by America's foremost manufacturers.



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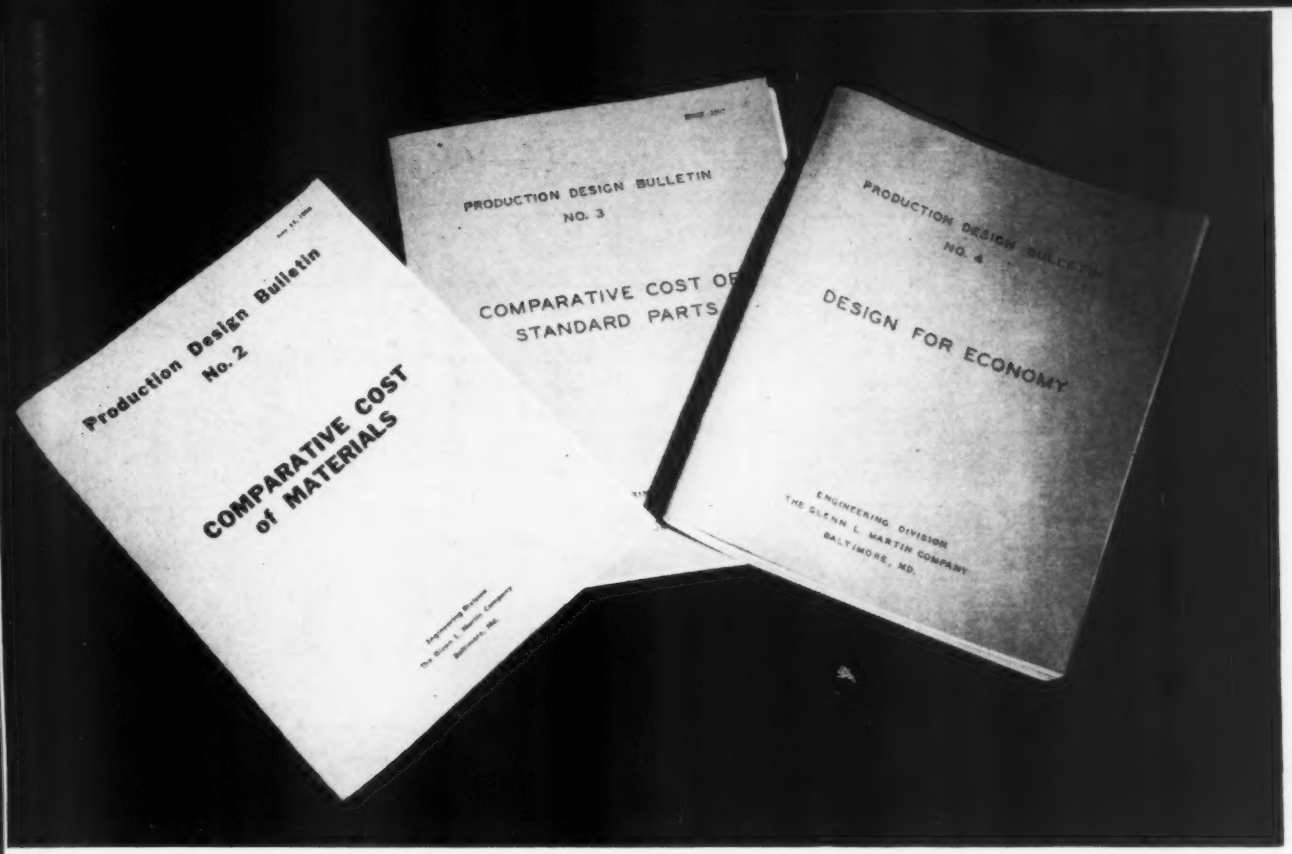


THREE HANDY REFERENCES containing cost data for engineering use help the engineer control expenses through product design.

Designing With Dollars

Brief analysis of elements of cost which determine the price of a product. Organization of cost data and its use by the design engineering department of one company.

JOHN VAN HAMERSVELD
Senior Cost Analyst, The Glenn L. Martin Co.

THREE ENGINEERING BULLETINS used by The Glenn L. Martin Company are helping engineers design better products for less cost. They provide engineers with up to the minute, well-organized cost data that can be used in the same way as other design information. This happy amalgam of dollars and sense is paying off not only in the engineering department but all the way down the line.

The cost data given in these bulletins has been prepared with due regard for all the elements that make up the cost of the finished product. In manufacturing, the main direct and indirect costs are:

DIRECT COSTS

Raw materials used in construction
Purchased or standard parts
Purchased or standard equipment
Labor

INDIRECT COSTS

Raw materials used in tools
Supplies
Machinery
Power
Labor

In preparing the bulletins, the cost control group of the engineering department set out to show how these costs were influenced by design. Then they prepared the information they worked up in several ways so that it could be easily understood and used by design engineers.

As a result of studies made by the cost control group, data was collected, organized and presented to the engineering department in three bulletins —(1) Comparative Cost of Materials, (2) Comparative Cost of Standard Parts, (3) Design for Economy.

COMPARATIVE COST OF MATERIALS

This bulletin contains cost data on all metallic materials used in the fabrication of parts. It covers a range of material items such as: Bar stock, sheet stock, tubing, extrusions, castings and forgings, in all commonly used alloys of aluminum, steel and magnesium.

All material cost data are shown in useful graphical form. For instance, the cost of sheet metal purchased in quantities from 100 to 300 lb is plotted showing the cost, in cents per square foot, versus sheet gage. Tubing cost data is given by showing the cost of tubing in cents per pound when tube diameter, gage and material vary. Data on sand castings, forgings and die castings are given in terms of cost per pound versus number of pounds

Table I—Percentage of Component Costs Controllable by Aircraft Designer

	Component Costs, Percent	Percent Controllable by Design
Raw Materials ...	11	5.50
Equipment	29	7.25
Direct Labor	15	6.00
Indirect Labor	15	7.50
Tooling	8	2.40
Engineering	7	2.10
Sales, Adm, Profit	15	...
	100	30.75

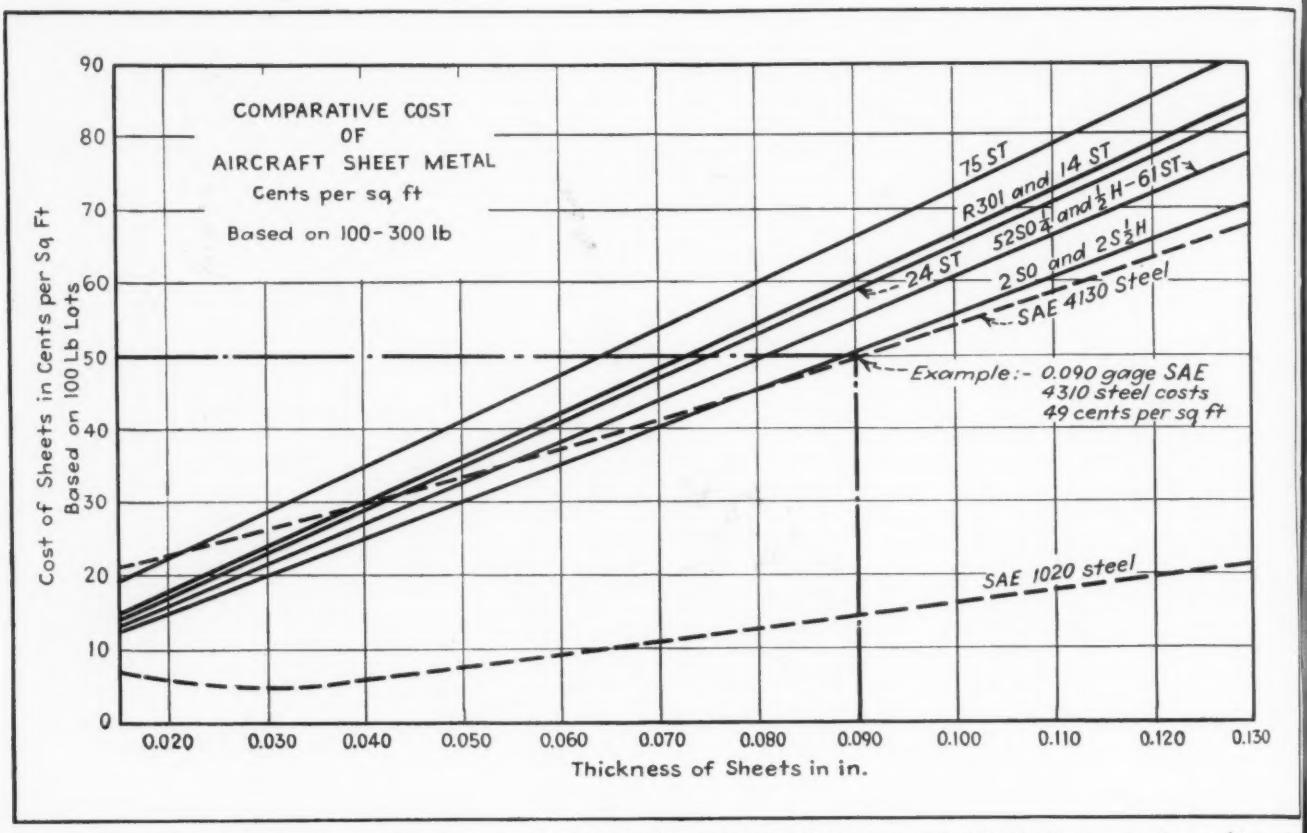


Fig. 1—Costs of various structural metals in terms of sheet thickness. Similar charts are prepared for forgings, extrusions, tubing, etc.

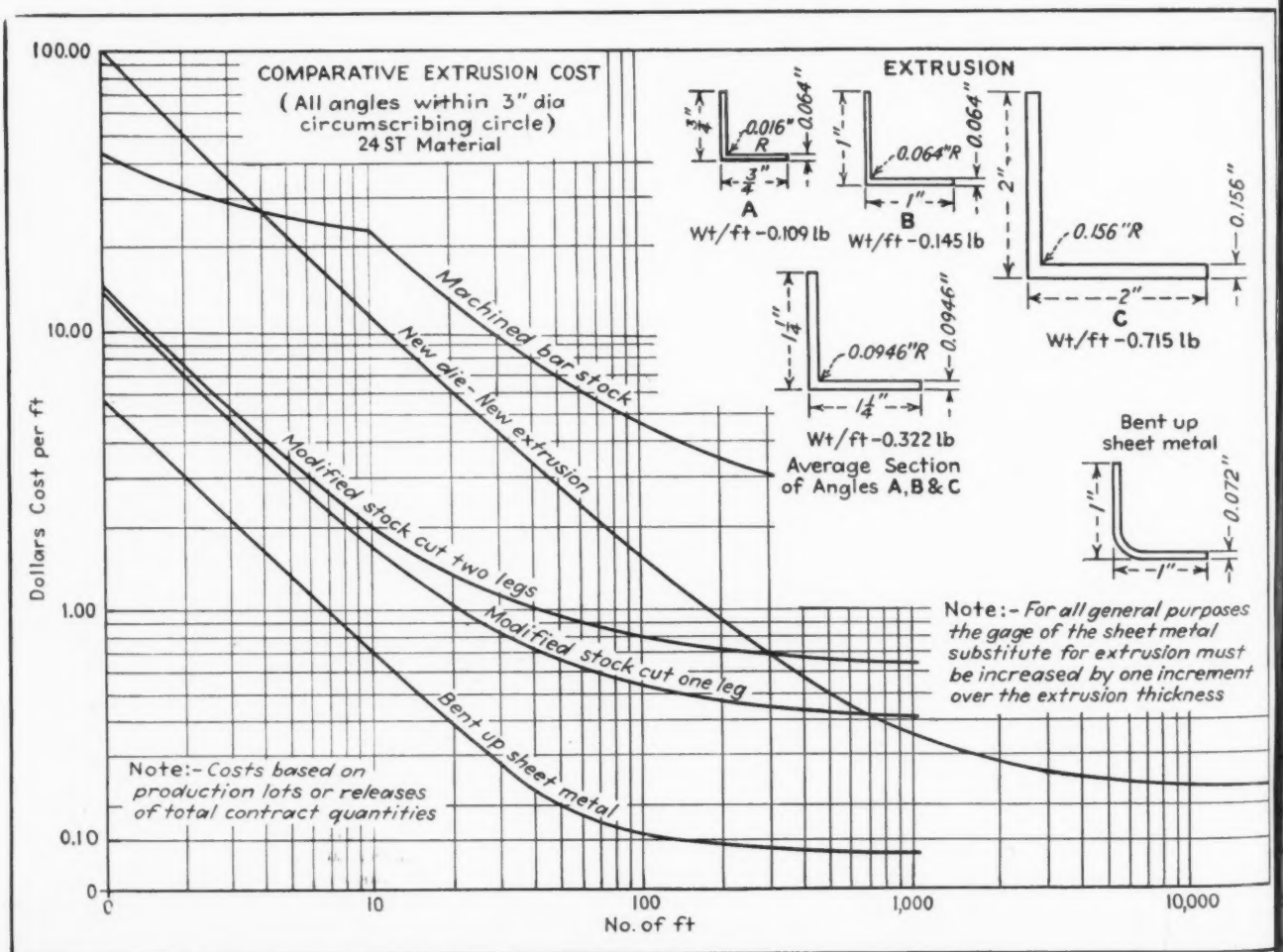


Fig. 2—Comparative costs for several types of structural angle used in moderately large quantities in aircraft construction.

per part for complex parts, normal parts and simple parts. Fig. 1 shows a typical material cost data sheet for aircraft sheet metal.

This type of data is adequate for comparing material as received, but must be supplemented with data showing the effect of shop operations performed on material. The following illustrates this statement. Aircraft designers frequently use structural angles for stringers, reinforcements, clips, etc. These parts may be made from extrusions, bent-up sheet metal, machined bar stock or modified extrusions. The cost of the finished part depends upon quantity of material required, length of each part, cost of dies, cost of machining, and cost of sheet metal operations.

To help the designer make an economical selection, a series of curves were prepared showing the cost of structural angles in terms of production methods and number of feet per production run. To prepare the curves, shown in Fig. 2, the cost control group did the following. First, from previous purchase records, they drew up a list of frequently used angles. By listing angle dimensions they found that the bulk of the angles had dimensions which fell within a 3-in. circumscribed circle. Second, they selected average sizes for extruded, machined and sheet metal angles. Finally they used the selected sizes to compute production costs for angle made from:

Machined bar stock
New die-new extrusion
Modified stock with both legs machined
Modified stock with one leg machined
Bent up sheet metal

In the same way, material cost data were prepared for other raw material items, which are modified in the shop before fabrication into specific parts or assemblies.

Material cost data given in the bulletin can be used easily. After the designer has established a satisfactory functional design, he determines the amount of material needed to manufacture the design. Using the material cost tables and charts he selects the most economical material to use. Depending on his judgment of the design factors involved, he can specify a forging at a dollar a pound or a sheet metal fabricated design at fifty cents a pound.

Comparative Costs Of Standard Parts

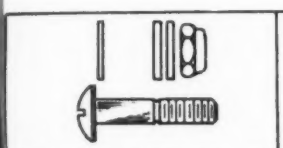
The second bulletin lists cost of standard parts such as; rivets, bolts, screws, and studs. Almost every manufacturing company uses standard parts in the fabrication of a product and of these, there are many sizes, types, etc. Usually the installed cost of a standard part rather than initial cost determines whether it can be used or not. Therefore, the cost control group collected and organized into two main divisions, complete information on the cost of all installed standard parts.

Part 1—Installed cost of standard parts used in general applications of structural designs

Part 2—Installed cost of standard parts used in general applications of non-structural designs.

All standard parts are listed by their specific commercial or AN Standard name, number, size, length, grip, weight and installed cost. In the total cost are the cost of the part, the labor cost for handling, drilling, reaming, counter-sinking, dimpling, riveting, deburring, tightening, checking by operator, and all other operations required to install the standard part. The time to perform each operation was evaluated from standard data and converted to dollar costs by applying proper fatigue allowances, take-away rates, learning cycle curve factor and overhead rates. The material costs for each standard part was based on general bulk shipment normally procured by the purchasing department for a number of production orders.

Comparing the cost of similar standard parts, as shown for a typical page in Fig. 3, the designer is able to select the most economical standard which meets design requirements. Where many types of bolts can be used to connect the fitting to a structural member, the designer has data



(1) AN960DL WASHER (AL. AL.)
(2) AN960D WASHERS (AL. AL.)
(1) AN364 NUT (STEEL)
(1) AN CLEVIS BOLT (STEEL)

DIA.	LGTH NO.	GRIP (IN.)	WT. (LBS.)	INST'D. COST (DOL.)
10-32	-9	1/4	.0110	.3710
	-13	1/2	.0129	.3791
	-17	3/4	.0149	.3903
1/4-28	-13	1/2	.0229	.3779
	-17	3/4	.0264	.3872
	-21	1	.0299	.3991
3/8-24	-17	3/4	.0561	.4566
	-21	1	.0631	.4825
	-29	1-1/2	.0781	.5032

(1) AN960DL WASHER (AL.AL.)
(2) AN960D WASHERS (AL.AL.)
(1) AN320 CASTLE NUT (STL.)
(1)AN381 COTTER PIN (CORR.STL)
(1)AN CLEVIS BOLT (STEEL)

DIA.	LGTH NO.	GRIP (IN.)	WT. (LBS)	INST'D. COST (DOL.)
10-32	-9	1/4	.0112	.8721
	-13	1/2	.0131	.8802
	-17	3/4	.0151	.8905
1/4-28	-13	1/2	.0228	.8885
	-17	3/4	.0263	.8981
	-21	1	.0298	.9077
3/8-24	-17	3/4	.0571	.9445
	-21	1	.0641	.9568
	-29	1-1/2	.0791	.9837

(1) AN960DL WASHER (AL.AL.)
(2) AN960D WASHERS (AL.AL.)
(1) AN365 NUT (STEEL)
(1) AN HEX. HD. BOLT (STL.)

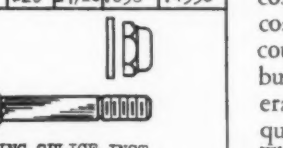
DIA.	LGTH NO.	GRIP (IN.)	WT. (LBS.)	INST'D COST (DOL.)
10-32	-5	1/4	.0146	.3174
	-7	1/2	.0165	.3291
	-11	3/4	.0184	.3380
1/4-28	-7	7/16	.029	.3292
	-11	11/16	.032	.3387
	-13	15/16	.036	.3490
3/8-24	-12	11/16	.075	.4059
	-14	15/16	.083	.4306
	-20	1-7/16	.098	.4556

(1) AN960DL WASHER (AL.AL.)
(2) AN960D WASHERS (AL.AL.)
(1) AN310 NUT (STEEL)
(1)AN381COTTER PIN (CORR.STL)
(1) AN HEX. HD. BOLT (STL.)

DIA.	LGTH NO.	GRIP (IN.)	WT. (LBS.)	INST'D. COST (DOL.)
10-32	-5	1/4	.0141	.8478
	-7	1/2	.0160	.8561
	-11	3/4	.0169	.8668
1/4-28	-7	7/16	.0270	.8629
	-11	11/16	.0300	.8733
	-13	15/16	.0340	.8845
3/8-24	-12	11/16	.0728	.9150
	-14	15/16	.0808	.9262
	-20	1-7/16	.0958	.9531

P4M-1 WING SPLICE INST.
(1)NAS143-8C WASHER (STL.)
(2)NAS143-8 WASHERS (STL.)
(1)INTERNAL WRENCH NUT (STL
(1)INTERNAL WRENCHING BOLT (UAB) (STL.)

DIA.	LGTH NO.	GRIP (IN)	WT. (LBS)	INST'D COST (DOL)
1/2-20	-28	.938	.2270	2.297



202 WING SPLICE INST.
(1) AN960D WASHER (AL.AL.)
(1) AN364 NUT (STL.)
(1) 26-2-9 SHEAR BOLT (STL)

DIA.	LGTH NO.	GRIP (IN)	WT. (LBS)	INST'D COST (DOL)
9/16	-22A	15/16	.1404	1.066
	-26A	1-3/16	.1594	1.084
	-28A	1-5/16	.1674	1.222
	-31A	1-1/2	.1814	1.257
	-33A	1-5/8	.1904	1.283

Fig. 3—Complete installed cost for various types and sizes of standard bolted attachments.

Table II—Design Check List As an Aid in Reducing Costs

Basic Design of Major Subassemblies

1. Can any of the parts be combined and made in one piece?
2. Can right- and left-hand parts be made identical?
3. Are cutouts and clearances adequate to insert and remove bolts easily?
4. Can curves and bends in the structure be reduced to simplify manufacturing?
5. Can simple mechanical devices be substituted for costly precision parts such as oleos, vacuum cylinders, etc?
6. Is there any way of reducing the number of separate parts in the assembly?
7. Are clearances between parts of subassemblies kept as large as possible?

Cost of Joining Sheets
Riveting, Stitching, Welding, etc.

8. Could stitching possibly be used in place of any other method?
9. Is riveting open and accessible for Erco automatic punch and riveting machine?
10. Can squeezers be used or must rivets be bucked with a two-man crew?
11. Can blind rivets be avoided by hand holes, cutouts, etc.?
12. If spotwelding is specified, do number of inches of weld justify cleaning cost?
13. Can other designs be substituted for arc or gas welded steel parts?
14. Could lap joints be substituted for butt joints to avoid close tolerances on trimming?
15. Can brazier head rivets be substituted for flush rivets?
16. Can parts be furnace-brazed together as a substitute for welding?

Cost of Assembly of Parts, Screws, Bolts, Etc.

17. Where fairings are assembled to ship, can slotted holes be used to decrease fitting time?
18. On forgings welded into a welded structure, has plenty of material been left for machining after welding?
19. Can lacing of fabric be replaced by self-tapping screws and retainer strips?
20. On removable parts, can attaching fittings be designed so that bolt or nut is "Floating" to take care of minor variations in parts?
21. Are electrical wiring harnesses in each major assembly joined to other assemblies by multiple plugs?
22. Can the swaging process be used in attaching fittings?
23. Can flash welding be used in attaching fittings?
24. Can elastic stop nuts, tinnerman nuts, etc., be used in place of safetying with cotter keys and wires?
25. Can fitting, trimming, filing, etc., required in the assembly operation be avoided by allowing for variations by lap joints and adjustments?
26. Are limits controlled so that interchangeability of major assemblies can be maintained?
27. In specifying AN fittings, has every effort been made to reduce the number of different parts by standardizing and limiting the list for the model?

Fabrication Costs

28. On compound curved parts, is it practical to redesign so that the part can be "wrapped" from flat sheet to avoid forming?
29. If wrapping is impossible, can parts be designed so that Erco stretcher can be used instead of the drop hammer?
30. Are all small clips and gadgets on airplanes standardized to a few basic sizes?
31. Are formed parts designed so that their manufacturing is practical on existing equipment without hand work?
32. Can parts be designed with flanges in one direction so that it can be formed in one operation?
33. Are tolerances liberal enough for limits of available equipment?
34. Do tolerances specified on machined parts need to be as close as what is shown?
35. Where pins are undercut or castings bored out to save weight are tolerances extremely liberal?
36. Can design of part be adjusted to utilize standard widths of aluminum, plywood, etc.?
37. Can parts be designed so that standard dies can be used instead of special dies?
38. Can standard cutting tools be used so that special reamers, milling cutters need not be purchased?
39. In dimensioning turned parts to fit in reamed holes, has standard reamer size been considered for the hole?
40. Can pilot holes in skins and extrusions be lofted to reduce jig cost and assembly time?

Substitutions to Reduce Cost

41. Can stock items such as standard channels, extrusions, rolled sections and tubes be used in place of special designs?
42. Can extrusions be substituted for forgings or castings in bolted fitting design?
43. Can purchased parts made in quantity for other industries be substituted for manufactured parts at a savings?
44. Can a cheaper material be used instead of the one that was specified?
45. Can 1020 steel be substituted for 4130 without undue reduction in strength?
46. Can 61ST be used instead of 24ST on parts requiring deep draw or stretch?
47. Can castings be made of "secondary metal" where they are non-structural?
48. Can standard bolts be substituted for special bolts and threaded parts?

Table III—Production Factors Related to Simplified Design

Simplification

1. Simplified design reduces interdepartmental work; fewer parts mean fewer drawings, lighter structure, less stock handling, fewer records, less space required and lower cost.

Practicability of Assemblies and Installations

1. Practicability is determined by availability, capacity, and types of tools used in production.

Avoiding Close Limits and Clearances

1. Tolerances should be liberal, consistent with part function.
2. Too-small tolerances may set up impossible requirements for shop or lead to high part cost.
3. Study individual part tolerances to get liberal limits without exceeding allowable assembly tolerances. If necessary rearrange dimensions.
4. If tolerances less than 0.001 are specified, ask, "Would I scrap piece if part came from shop 0.001, 0.002, 0.003, or even 0.005 in. beyond limits I have set?"
5. Use standard tolerances if they exist. Always check shop tolerances for reamed holes, drilled holes.

Use of Existing Parts and Standards

1. Keep inventory low; use available stock if you have a choice.
2. Avoid specifying new materials and processes, unless they have been tested for production performance and cost.
3. Avoid expensive tooling; use existing dies or parts. To design calls for gussets, clips, brackets or machine fittings, check existing parts in current production.

COMPARATIVE COST SUMMARY SHEET MODEL: P4M-1

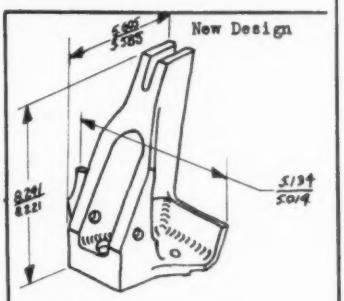
TITLE: Fitting - Lower Nacelle Longeron Forward

DRAWING NO.: 21-5052 049

ANALYSIS: Machined 14ST Forged Bar VS Machined Aluminum Forging VS 4130 Steel Welded Fitting H.T. 150,000 p.s.i.

PREMISES: 7 ships - 4 parts/ship - Total 28 parts 1 shop lot.

Tool and labor hours were coordinated with representatives from Tool Engineering and Detail Manufacturing.



Mat'l. 14ST
Specially Processed Bar

SUMMARY:

ITEM	BAR STOCK		FORGING		WELDED		CASTING CLASS	
	TOTAL COST	UNIT COST	TOTAL COST	UNIT COST	TOTAL COST	UNIT COST	TOTAL COST	UNIT COST
FIXED TOOLS					▲ 124.54	4.44		
MAT'L		20.79		10.50		5.52		
SET-UP	110.06	3.91	36.12	1.28	56.84	2.03		
LABOR		40.88		10.40		20.96		
DIE			• 2000.00	71.43				
DIE SET-UP			• 30.00	1.07				
ATTERN								
ENG. COST					131.04	4.68		
SPEC. TEST								
TOTAL UNIT COST		65.58		94.68		37.63		
UNIT WEIGHT	3.4 lbs.		4.00 lbs.		4.3 lbs.			

REMARKS: It is recommended the welded steel fitting be used for production ships.

▲ Includes Dev. Template for Detail #2 and Assem. and Weld fixture.

• Die Cost and Die Set-up cost amortized over 28 parts.

PREPARED: Thornton DATE: 8-26-47 APPROVED: [signature]

CHECKED: [signature] DATE: 8-29-47

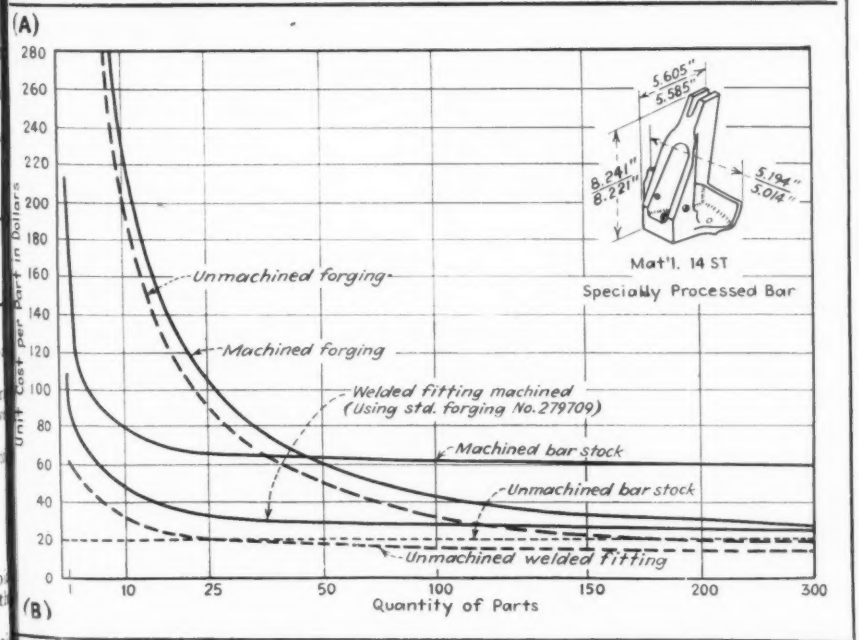


Fig. 4—Cost study for one fitting, (A) comparative cost summary sheet, (B) study expanded to show cost for other quantities. Note changes in relative costs with quantity.

on several types of standard bolts at his finger tips. His selection sets the cost of installation for the fitting. Knowing the cost, strength and weight of the selected standard part, he can discuss his selection with the stress and weight engineers for further confirmation. Shown in this bulletin are installed cost data for Hi-shear rivets compared with bolts, blind rivets compared with regular types, and Airloc fasteners compared with Anchor Nuts and many other combinations of standard parts.

Standard parts cost data can be used for general estimating also.

Design for Economy

The third and last bulletin "Design for Economy" contains information that is hardest to assemble from the standpoint of the cost control group. The object of the bulletin is to imbue the designer with a sense of design prospective. Although cost data given in the first two bulletins are easy to use, they can be considered only as a supplemental aid to the designer. The art of design cannot be reduced to rules because it derives from the sum total of the designer's experience and from the degree of intelligence with which he applies his knowledge to current problems and situations.

Because of this, the third bulletin avoids a too-detailed treatment of design details. The first part is a design check list. Part of the list appears in Table II. The questions listed are the sort a design checker might ask in reviewing a drawing.

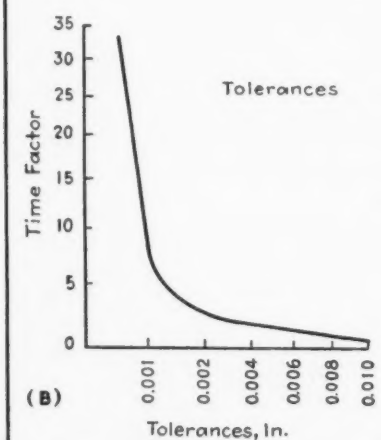
The second part of the bulletin discusses several production factors such as: Simplification, practicability of assemblies and installations, limits and clearances, and use of existing parts and standards. Some pertinent remarks appearing in the discussion under each of these headings are given in Table III.

The remainder of the bulletin is a comparative cost analysis of classified aircraft fittings. The data is given in cost summary sheets. One of these sheets is shown in Fig. 4(*A*). The data is further expanded as shown in Fig. 4(*B*). The summary sheet shows the complete cost breakdown for a fitting fabricated in several ways and includes the cost of fixed tools, materials, set-up, labor, die, die set-up, pattern, engineering labor, and specification testing.

A tabulated cost summary, for all typical fittings analyzed, is inserted in the bulletin to emphasize the fact that weight and manufacturing cost

Surface Finishes	
Rms Value $\left(\frac{\text{000}}{\text{}}\right)$	Approx. Relative Cost
500	1
250	2
100	5
20	15
5	50

(A)



Note:- More parts per man hour can be made by "opening up" machining tolerances to 0.002 in. or more

Fig. 6—(A) Correlation of relative costs with surface finish (B) Correlation of relative machining time with tolerances.

are closely related. As pointed out in Fig 5, cost studies show that a small increase in weight accompanies a reduction in production costs. Here again the designer must take the initiative in properly evaluating the application of the cost data shown to his specific design problem.

One additional insert in the design for economy bulletin can be mentioned. This is the designer's "A B C" check list of questions and reminders. These are listed as follows:

A—Before designing a part, ask

1. Why you need it—can you cancel it
2. When is it used—how long must it last
3. What specifications must be met
4. Where does it go
5. Is it easy to install
6. Is it easy to use
7. Is it easy to service

B—During design, ask

1. How can it be combined with another part
2. How can it be simplified

C—Try to design for

1. A minimum number of simple members
2. Straight members, unless forming simplifies
3. Standard parts, shapes, sizes, dies
4. Maximum satisfactory tolerances and finishes
5. Greater than minimum bends, clearances, etc.
6. Low cost materials, for example SAE 1020 Steel vs SAE 4130 Steel
7. Forming aluminum alloy in ST condition
8. Minimum of low cost fasteners
9. No closed angles or re-entrant cuts
10. Fabricate part in shop on available equipment, rather than subcontract

D—Regularly consult your

1. Drafting room manual
2. Cost bulletins
3. Cost control group

The foregoing list may seem redundant and trite, but it does touch upon a number of design points which, if overlooked or ignored, result in added cost due to rework.

The accumulation of cost data as described takes time and money and can be done only by experienced design personnel thoroughly familiar with the company's products, production methods and production equipment. Also such cost data must be current to be of any value. But there is no doubt that it is a good investment; particularly when previous studies have shown that designers have control over approximately thirty percent of the production cost of an airplane. The dollar sign can be as important a tool as a French curve.

TABULATED COST SUMMARY

STUDY NO.	TITLE	PIECE WEIGHT DIFF. +	PIECE WEIGHT DIFF. -	QTY. SHIP	TOTAL QTY.	PROTO. DESIGN	PROPOSED PROD. DESIGN	SHIP WEIGHT DIFF. +	SHIP WEIGHT DIFF. -	PIECE COST DIFF. +	PIECE COST DIFF. -	CONTRACT COST DIFF. +	CONTRACT COST DIFF. -
* 1	Ftg. Center Wing Jet Mt.	+ .045		4	28	Bar	Weld	+ .180			-$ 4.45		-$124.60
2	Link Ass'y Center Wing	± 0		12	84	Bar	Bar (OK)	± 0		± 0		± 0	
* 3	Ftg. Oleo Door (Male)	+ .208		2	14	Bar	Weld	+ .416			-$ 4.19		-$ 58.66
* 18	Handle Receptacle	+ .116		2	14	Bar	Casting (Class B)	+ .232			-$ 1.56		-$ 19.04
19	Handle	± 0		2	14	Bar	Bar(OK)	± 0		± 0		± 0	
20	Main Landing Gear Ftg.	± 0		2	14	Bar	Bar (OK)	± 0		± 0		± 0	

** May be reduced 40% by careful Stress Analysis

* TOTAL SUMMARY OF 9 STUDIES

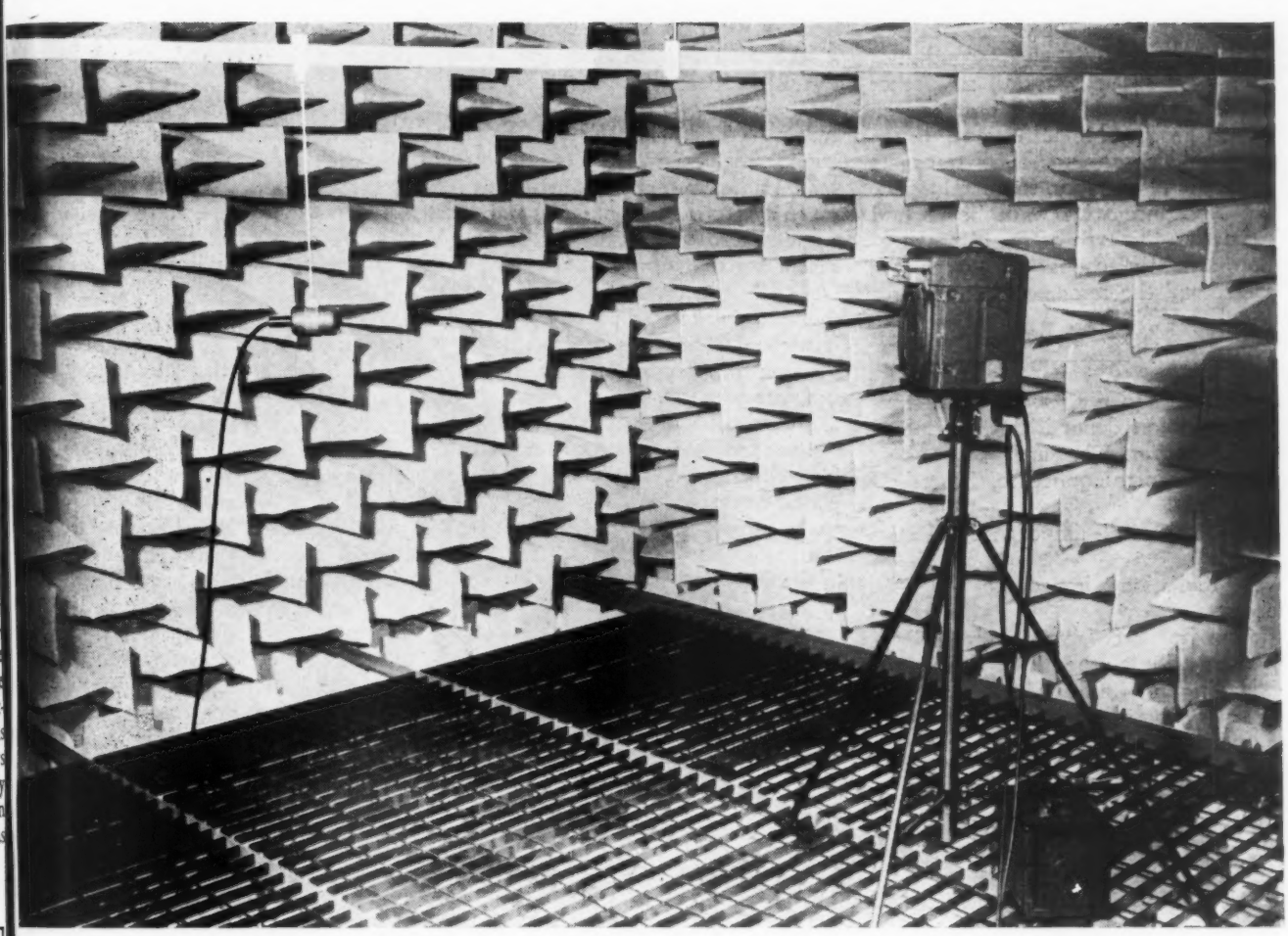
Total cost savings per contract ≙ $6,222.22
Weight Difference per ship = + 23.84 LBS.
Total cost savings per ship $6,222.22 / 7 = $ 888.89

For every lb. increase in weight it will save G.L.M. Co. $888.89 / 23.84 = $ 37.70

What weight allowance has been made for Production Design?
It will cost G.L.M. Co. $37.70 to save 1 lb. per ship. This figure is in excess of $20/#, therefore, redesigns for production are recommended.

^ If Static Test is required by Navy, this figure must be modified.

Fig. 5—Tabulated cost summary sheet for twenty fittings show large savings possible by redesign with little increase in weight.



ANECHOIC (ECHOLESS) ROOM. Walls are constructed of rectanguar blocks of fiberglas, cut on the diagonal, then fastened back to back. Each wedge is enclosed in a cheesecloth stocking. In this set-up, noise level of some automobile equipment is being measured.

Sources of Machine Noise

- How sound energy is created and transmitted.
- Mechanical impedance, radiation impedance and the effect of simple and dipole radiators in noise reduction.
- Technique of sound measurement, including cost of minimum and complete equipment for an acoustic laboratory.
- Procedure for analyzing and reducing machine noise.

HOWARD C. HARDY

Supervisor, Acoustics and Vibrations Section
Armour Research Foundation of Illinois Institute of Technology

AUDIBLE SOUND is produced when a moving part causes air pressure variations with component frequencies that can be detected by the ear. A misconception is that the pressure variations must be sinusoidal if the sounds are to be heard. Only the simplest sustained musical tones, the hum or whine of motors and the vowels of speech approach true sinusoidal character. Most sounds originate from impact, friction and other causes that create a pulselike wave front. These excitations, which may occur at random time or at a constant repeated interval, produce a brief pressure variation. Mathematically analyzed, such a pulse has a broad band spectrum, exciting all frequencies. The exact spectrum depends on the shape of the pulse.

If most sound energy sources are the impulse type, what gives the characteristic tones of such sound-radiating objects? These tones are caused by resonant frequencies excited by the energy source. For instance, when a bell is struck, certain portions of the background spectrum are amplified and vibrations are transmitted to the air, thus giving the characteristic tone or resonance. When the resonance is great and the tonal relationship of the resonant frequencies is harmonic, the sound may be musical in character. The more common annoying sounds are usually less resonant and have frequency components that are not harmonic.

The resonating parts do not have to be near the source of sound energy to cause some objectionable noise. Many examples of this occur in the

automobile where squeaks and rattles, which are usually resonant conditions in panels, frame shackles, and other parts, are excited by impact of the tires against road imperfections. In the household a running faucet transmits water noise via pipes, frequently causing resonance with wash bowls and basins. In this type of sound propagation it is the coupling between the source and the resonator that requires attention.

Factors Affecting Amplitude

The amplitude of sound from a vibrating object depends on three physical properties. They are:

1. The force applied to the radiating object.
2. Mechanical impedance of the object.
3. Radiation impedance of the object.

The first two factors control vibration amplitude, the third determines how this amplitude is transmitted to the air. The applied force can be reduced by decreasing the energy of the exciting source and reducing the efficiency of the mechanical coupling between the source and the radiating object. This important subject will be discussed later on.

The mechanical impedance, Z, is defined, where F is the alternating force applied, A is the amplitude, and f is the frequency, by

$$A = \frac{F}{2\pi f Z} \qquad (1)$$

In simplest analysis, impedance is a function of the effective mass M of the object, the effective stiffness K, and the damping resistance R. These are related by an expression analogous to electrical impedance,

$$Z = \left[R^2 + \left(2\pi f M - \frac{K}{2\pi f}\right)^2\right]^{1/2} \qquad (2)$$

Proper units for each of these terms are in the metric system:

Z or R — mechanical ohms' dynes sec cm^{-1} or gram sec^{-1}
M — grams
K — dynes cm^{-1}

It follows that the resonant frequency, f_o, occurs when the mass and stiffness components cancel:

$$2\pi f_o M = \frac{K}{2\pi f_o} \qquad (3)$$

$$f_o = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \qquad (4)$$

The mechanical impedance then has its smallest value and therefore the amplitude is at a maximum, its size controlled only by the frictional term R. This condition should be avoided.

When R is extremely large, the motion is called "resistance controlled." Increasing the friction may be a practical method of reducing vibration only if excessive friction does not impair the useful performance. If either the term containing K or M is the large factor, which depends on the frequency region involved, the object is said to be, respectively, stiffness or mass controlled. For most materials the average device is mass controlled except for vibrations at extremely low frequency (below 50 cps). One way to reduce sound amplitude, but only if the object is mass controlled, is to increase the effective mass of the moving part or parts.

If the sound is from an energy source that has a prominent exciting frequency as, for instance, a motor or generator running at 3,600 rpm (60 cps), the supports must be such that the modes of vibration are not equal to the frequencies of the exciting force or multiples of it. In these troublesome instances a slight change of mass or stiffness in the supports may change the resonant characteristics enough to remove the difficulty.

This discussion on mechanical impedance has been over-simplified. In actual design it is difficult and generally unnecessary to know the exact values of R, K and M. Almost always there are many resonant frequencies. The engineer should understand the relationship between these quantities so design changes can be made in the right direction. For a better understanding of R, K and M refer to Den Hartog, "Mechanical Vibrations," McGraw-Hill (1947) and A. L. Kimball, "Vibration Prevention in Engineering," John Wiley & Sons.

Types of Resonators

It is convenient to classify resonators into three simple classes. These are (1) enclosed air spaces including tubes, (2) rods and bars, and (3) plates and membranes. Resonators such as (2) and (3) are too complicated to be described in detail here, but their troublesome effects must be prevented. For a summary of the mathematical expressions applying to the simplest cases see P. M. Morse, "Vibration and Sound," McGraw-Hill (1936).

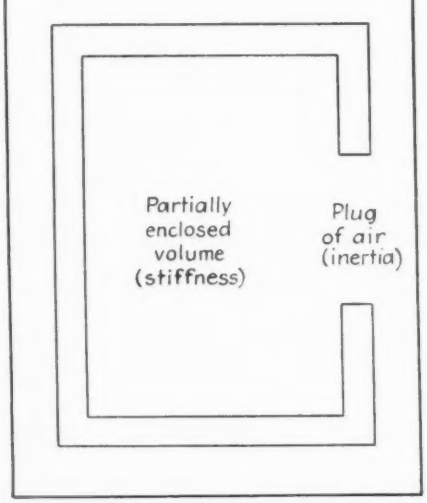


Fig. 1—Partially enclosed air spaces are often overlooked as a possible noise source.

Enclosed air spaces, however, are an important source of resonance often overlooked by the designer. Characteristics of such a system are given in Fig. 1. The enclosed volume provides a stiffness to compression and rarefaction of the sound wave. The small plug of air in the opening or openings has a definite inertia. At a certain frequency the stiffness of the partially enclosed volume and inertia of the air plug resonate with each other. Measurement, rather than calculation of the resonant frequency, is required because of the complicated geometry of most designs. Examples of such cavity resonators are motor housings, relay housings, oil burner combustion chambers and air conditioning plenum chambers.

If the sound energy source has frequency components near the frequency of the resonant chamber, the results definitely will be bad. Tuning the resonator to another frequency can be accomplished by changing the enclosed volume or the size of the opening. Increasing the interior surface area or covering it with absorbent material will increase the damping in the resonator.

Radiation Impedance

Although an object may have a large amplitude, it does not follow that it transmits much sound energy to the air. Effective coupling of the vibration with the air is controlled by the radiation impedance. Rigorous treatment of this subject requires much mathematics, but some principles can be expressed in a simple fashion.

A sharp distinction should be made between simple radiators of sound and dipole radiators. In the first, all the vibrating parts contribute wave motions in phase. Examples are a horn, a kettle drum, a gas engine exhaust, and a loudspeaker with the back enclosed. A dipole radiator has two radiating surfaces in opposite phase. Examples are a loudspeaker not enclosed in a box, a vibrating wire or tuning fork, a vibrating fan blade and a cymbal. In Fig. 2 the characteristics of simple and dipole radiators are compared.

The amount of sound radiated from

a simple source that has small dimensions, as compared to the wave length of the emitted sound, is about the same in all directions. A wave length for 100 cps is 11 ft, for 1000 cps, 1.1 feet. The amount of sound radiated from a dipole source is far less, because the radiations from the two sides are opposite in phase. Along the axis of the dipole radiator complete cancellation occurs.

The power radiated from a simple spherical source, having a radius a, where f is the frequency, ρ is the air density, A is the amplitude, and c is the velocity of sound, all in metric units, is

$$P_s = \frac{8\pi^3 f^4 \rho a^4 A^2}{10^7 c} \text{ watts} \qquad (5)$$

The same equation for a dipole, two sources of opposite phase separated by a distance a, is

$$P_d = \frac{8\pi^5 f^6 \rho a^6 A^2}{3 \times 10^7 c^3} \text{ watts} \qquad (6)$$

Thus the ratio of the two is

$$\frac{P_s}{P_d} = \frac{3c^2}{\pi^2 f^2 a^2} = \frac{3\lambda^2}{\pi^2 a^2} \qquad (7)$$

substituting the wave length λ for c/f. Thus, from a loudspeaker whose effective diameter is six inches, the ratio of the power radiated at 110 cycles ($\lambda = 10$ ft) from an enclosed speaker to one at the same amplitude but not enclosed is 485, or 27 decibels. This equation, being a ratio, is independent of the units used. The power input in both cases is the same, which illustrates how great an effect the radiation impedance can have on the efficiency of a sound source.

An excellent way of reducing noise is to reduce the radiation impedance. Usually the radiation impedance will vary as the fourth power of the linear dimensions if the emitting surface or hole is approximately square or round, and as the square of the width, if it is a narrow source. The radiation impedance also varies approximately as the square of frequency for a round or square source and as the first power of the frequency for a narrow source. It should be remembered, however, that the amplitude usually will be a function of the frequency. For instance, in mass controlled vibrations, the radiation efficiency is almost independent of frequency for objects smaller than a wave length.

It is important, therefore, to keep the size of the radiating object small. The amplitude of the vibration of a violin string is large compared to the violin itself. Yet a string vibrating alone cannot easily be heard. Most of the sound is radiated by the violin, not the string. An analogous case is the vibration of a flexible choke rod in an automobile. The radiation from the rod is negligible, but when it is coupled to an efficient radiator, such as the dash of the car, the sound output may be annoying. Similarly, a typewriter or a tuning fork coupled to a desk makes the top an effective sounding board.

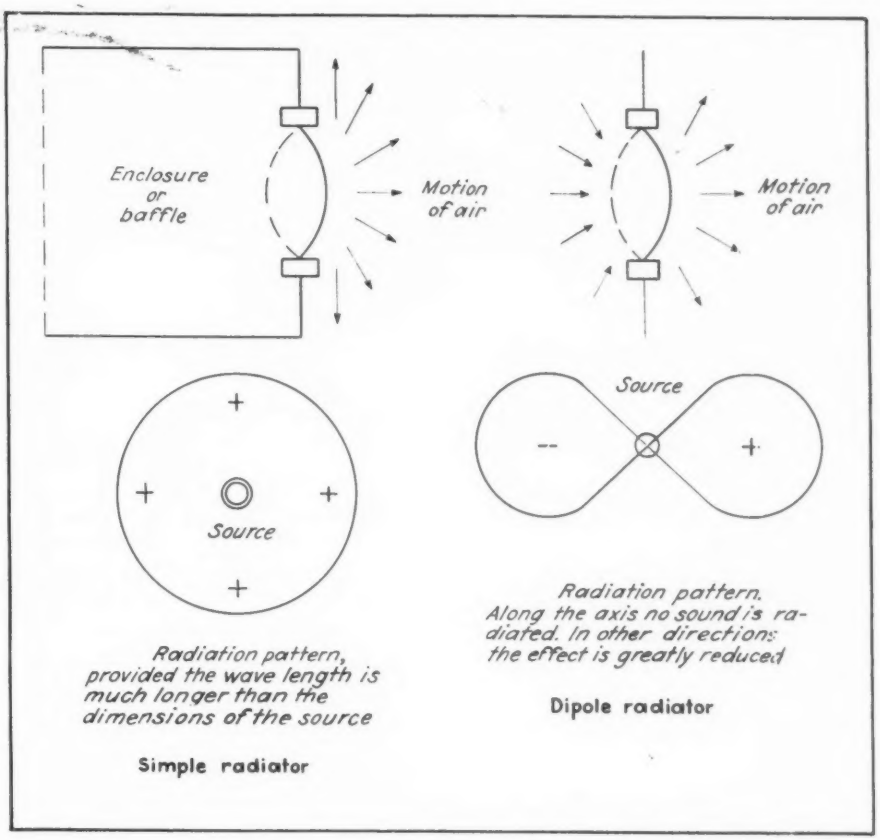

Enclosure or baffle
Motion of air
Motion of air
Source
Source
Radiation pattern, provided the wave length is much longer than the dimensions of the source
Simple radiator
Radiation pattern. Along the axis no sound is radiated. In other directions the effect is greatly reduced
Dipole radiator

Fig. 2—Simple and dipole sound radiators. By making a simple radiator into a dipole, noise radiation can be greatly reduced. This application is frequently possible in machinery.

Analyzing a Machine Noise

Questions that will determine the possibility of noise reduction by increasing mechanical impedance and reducing air radiation are as follows:

1. What is the source of sound energy?
2. What is resonating?
3. Is there more than one resonating element?
4. How is the source coupled to the resonators?
5. Is the vibrating part, or air cavity, controlled by mass, stiffness, or resistance?
6. What are the practical possibilities of increasing the mechanical impedance?
7. Will changing the resonant frequency of the resonator be an advantage?
8. Approximately what is the wave length of the most characteristic frequency?
9. What parts of the resonators, surfaces or holes, are radiating?
10. Are the radiating surfaces radiating in phase? That is, is the radiating surface a simple radiator, a dipole, or a combination of these?
11. Can the radiating surface be conveniently reduced?
12. Can the radiation be changed from a simple source to a dipole?

For the answer to some of these questions, which are not in any significant order, the engineer will have to use instruments.

Measurement of Sound

In sound measurements it is the physical properties that are measured, and not the sensation itself. When the physical data are obtained, they can be presented as such or converted into approximate sensation data by conversion charts as shown in "Fundamentals of Noise," Product Engineering, February, 1948.

For accurate observations on noise-producing equipment the system must be isolated from all other noise sources. Sometimes part of the machine must be isolated and driven by an outside means to avoid noise from other com-

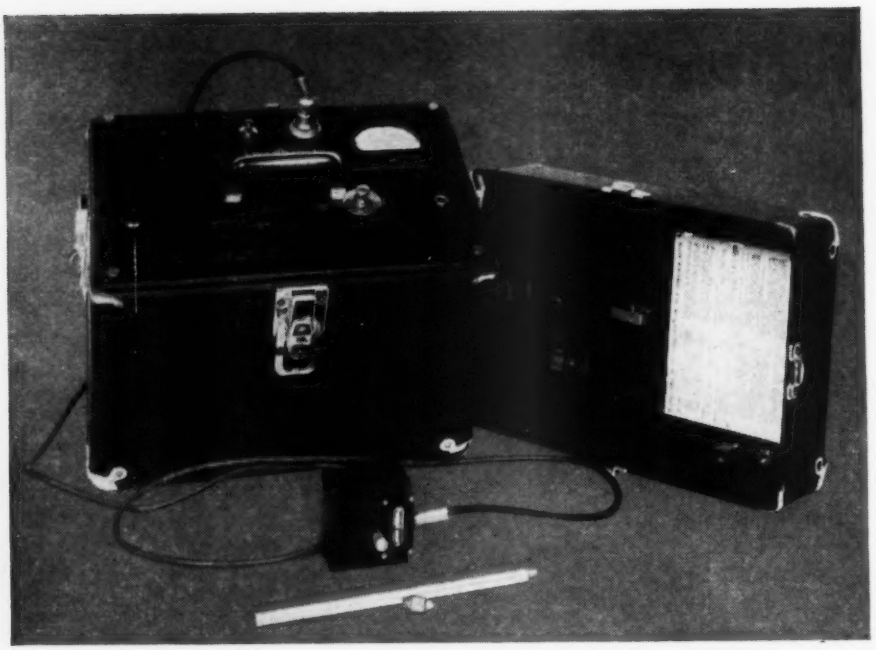


Fig. 3—Vibration meters are useful under 500 cps for determining the point of maximum amplitude on a large radiating surface. For high frequencies the sound must be measured.

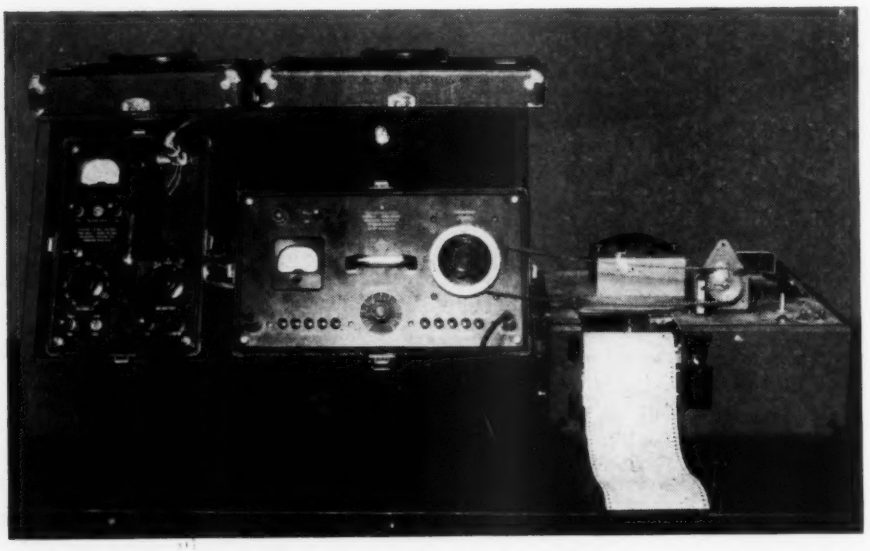


Fig. 4—Equipment to measure the spectrum of a noise: Sound level meter, sound analyzer, and recording meter. Analyzer frequency dial is turned by motor of recording meter.

ponents. The ordinary laboratory is sufficiently quiet for most observations, provided that the observer is not interested in precise intensity levels. Exact intensity levels, however, are seldom necessary. For accurate comparison of two sound sources or for absolute calibration, the reflections from walls, ceilings, floors, and furniture must be eliminated, and the measurements be made in an enechoic room.

The observations usually needed are, in order of importance: (1) source of the sound, (2) frequency spectrum and (3) relative intensity level. It is seldom necessary to measure the exact intensity level except for an occasional check on improvements. The loudness heard by the ear is usually as good a guide as any other.

The best research tool that an acoustic engineer has is his trained ear. It is rather easy to educate oneself to estimate sound levels on a meter by listening to a loudspeaker connected with an audio-frequency oscillator. An acoustical engineer should become so acquainted with the pitches of various sounds that he can estimate their approximate frequencies. When the noise spectrum contains many tones it is better to resort to instruments.

To find the actual radiating source of noise is often difficult, particularly when the sound is of low frequencies. One of the most efficient instruments for this purpose is the ordinary clinical stethoscope. This simple instrument is a great aid to the acoustic troubleshooter, although its use has generally been overlooked. Its chief advantage is in searching for the exact radiation source, and for the parts of the source where the noise is the worst. It is also useful for studying resonating air cavities.

Measuring the Spectrum

To obtain data on low frequency vibrations, a vibration meter, Fig. 3, is a useful instrument. On large surfaces the point of maximum amplitude can be observed. Since the amplitude is small at high frequencies, this type of measurement is only applicable to 500 cps frequencies or below. Measurement of the sound itself is the only practical method for high frequencies.

To measure the spectrum of the noise, the setup shown in Fig. 4 is convenient. A sound level meter picks up the sound through the microphone. The output of this meter is introduced into a sound analyzer. The amount of sound at each frequency can be read on the meter scale. Sound analyzer output can be introduced into a recording meter for a permanent record.

To assure that the spectrum recorded is the spectrum actually desired, a good technique is to place the microphone in a position where the ear has heard the unfavorable sounds. Readings from more than one position are usually advisable. The use of the sound spectrum in attacking a noise problem will be disscused in the third and last article of this series.

Measuring Intensity

The intensity level is obtained on a sound level meter. It has been pointed out that the relation between intensity and loudness is a complicated function of frequency and the number of frequency components. See Fig. 3, "Fundamentals of Noise," PRODUCT ENGINEERING, February, 1948. To partly adjust for the frequency effect, several networks are usually provided in the sound level meter. For instance, in the General Radio Type 759-B, there is a flat response network, a second network adjusted to get the loudness contour at 70 decibels, and a third at 40 decibels.

Sound intensity measurements are difficult because of reflections from walls and other objects. Sound level data may be useful, however, when two objects which have essentially the same noise spectrum are compared, and when the source and microphone are placed in turn in the same position in the room. Another use of the

sound level meter is to obtain quick readings of noise background. A good guide in the use of acoustical instruments is "The Noise Primer" by the Engineering Dept., General Radio Co. (1946).

Recording

Acoustic engineers have needed a convenient device that would record sounds for subsequent playback. Recent developments in magnetic recording are the answer. Magnetic recordings are particularly useful for a day-by-day notebook of progress. They also facilitate the recording of sounds that occur at rare intervals, thus enabling their analysis at leisure.

If the noises being studied have fast transients, such as the clicks of a relay, the meter readings may be too slow to catch the sound. Then it is best to study the sound on a cathode-ray oscillograph. With proper technique, the intensity, characteristic frequency, if any, and the decay time can be measured. If desired, the phenomena can be photographed.

Optics in Noise Reduction Work

Photographing oscillograph traces is one use of optics; another is the stroboscope. This instrument is useful when the noise-producing motion is repeated at regular intervals, as in rotating or reciprocating machinery.

High speed photography is valuable if the motion is of considerable amplitude, but not necessarily repetitive at a fast rate. Seeing the motion of the noise energy source is a great aid in prescribing a cure. The Western Electric Fastax camera is one of the best tools for this work.

Equipment for Sound Studies

At the upper right is a list of what one might call essential equipment for a laboratory studying noise problems.

The construction of an anechoic room, or at least a room with highly absorbing walls, will be necessary for elaborate experiments. Several soundproof booths for noise isolation should also be available.

The techniques of obtaining the measurements and the astuteness of the personnel who interpret the meter readings are far more important than the accuracy of the instruments. Experience in using the instruments will be developed easily, providing the engineer has a good acoustic background.

Since the psychological aspects of sound, the physical sources of sound and sound measurement have been discussed, the third and last article of this series will cover the application of this knowledge to particular problems.

Minimum Equipment For An Acoustic Laboratory

Stethoscope	$ 5
Sound level meter	320
Sound analyzer	400
Stroboscope	125

For the More Elaborate Laboratory the Following Additions are Suggested

Vibration meter	$400
Sound level response recorder	900
Cathode ray oscillograph	200
Magnetic recorder, wire or tape	200
Beat frequency oscillator	450
High quality microphone	200
Audio amplifiers	300
Electronic voltmeter	250

For the Complete Laboratory this Additional Equipment Should be Available

High fidelity magnetic recorder	$1,500
High speed level recorder	1,000
Vibration analyzer	475
Fastax camera	1,500
Good loudspeaker	300
Set of octave band filters with associated high quality amplifier	700

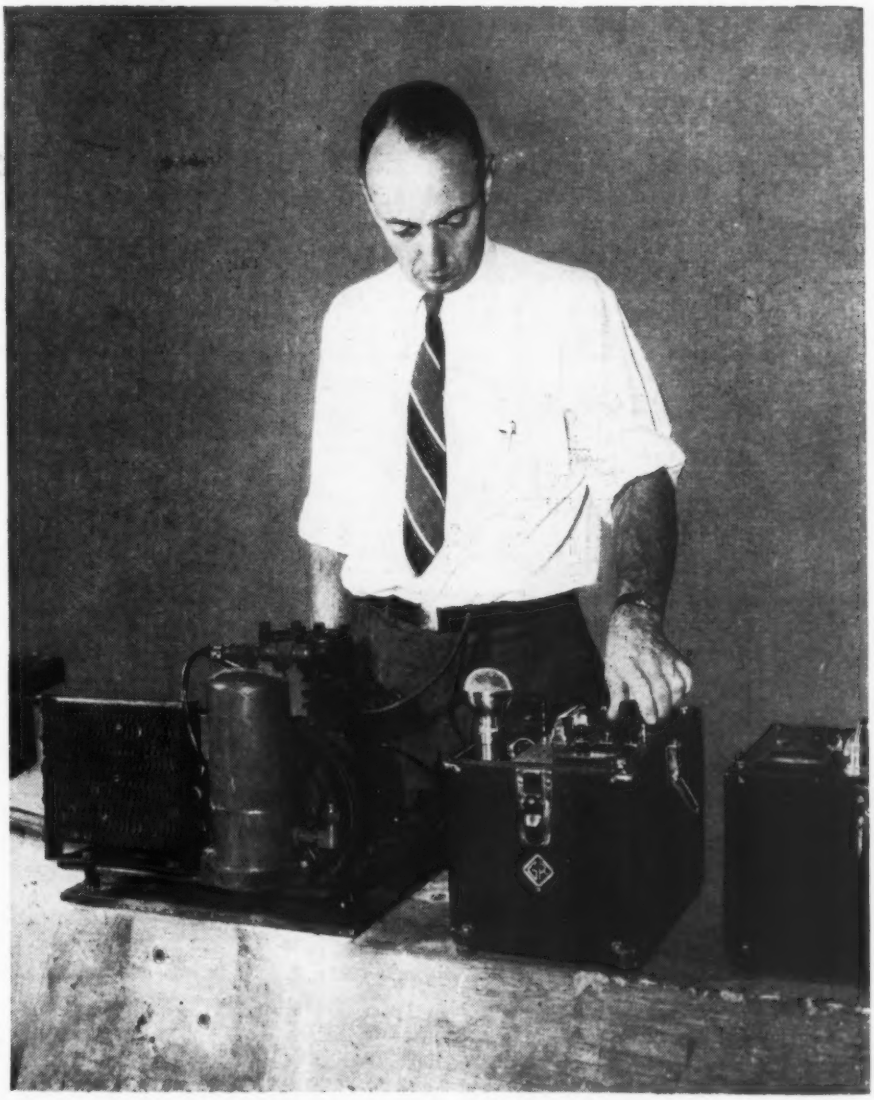
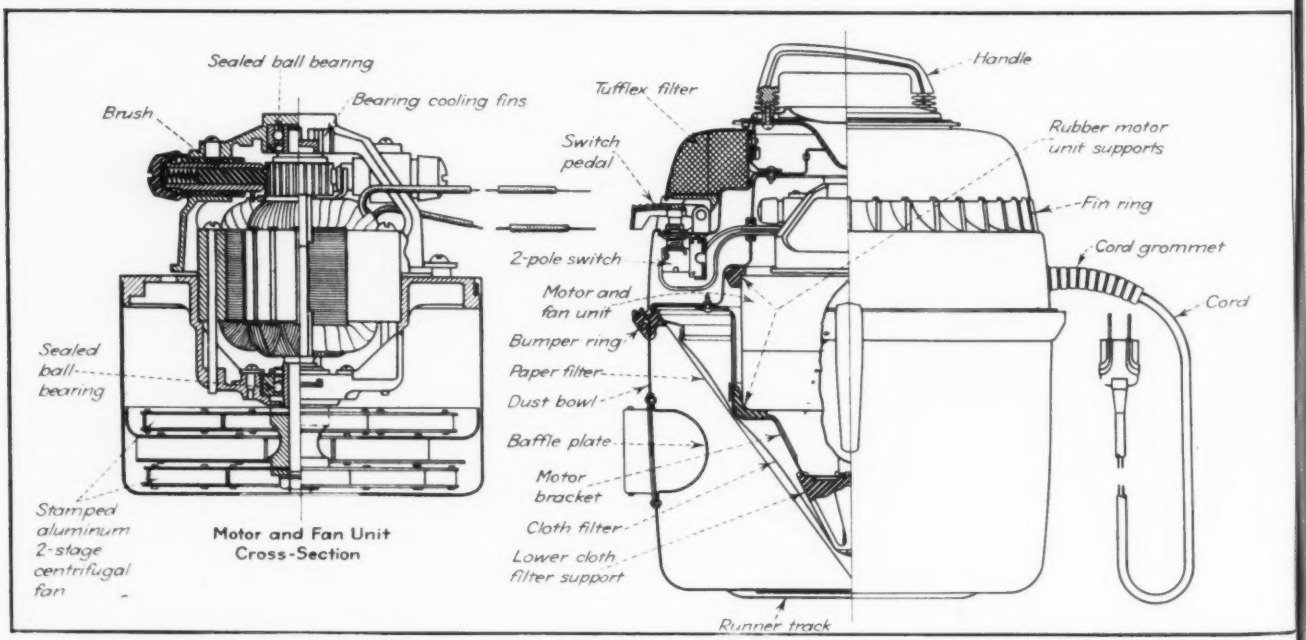


Fig. 5—Measuring the sound of a refrigerator compressor. A soundproof booth or even a quiet laboratory is adequate for such a test. An anechoic (echoless) room is seldom required.

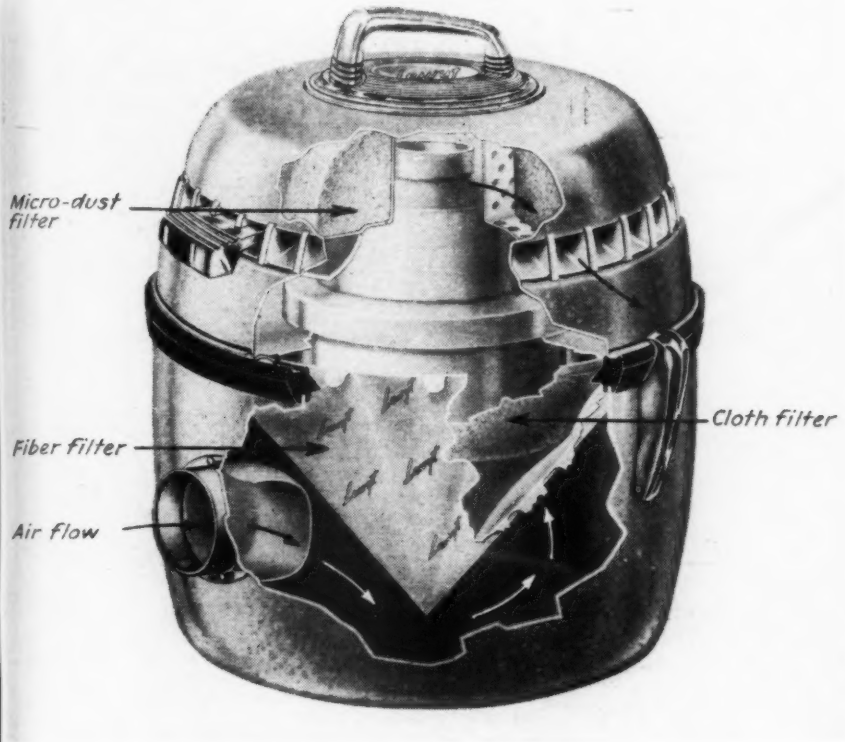
Vacuum Cleaner Designed for Quiet Operation

Quiet operation of this vacuum cleaner unit has been effected by exhausting the air through a finned diffuser having 36 openings equivalent to 20 square inches of area. Canister type cleaner unit is provided with two runner tracks, permitting it to be drawn along the floor by the hose that connects attachments to the unit. Cleaner unit weighs 15 lb 14 oz, is $14\frac{1}{8}$ in. high and has an $11\frac{1}{4}$ in. diameter. Attachments weigh 7 lb, 12 oz. Suction is furnished by a 2-stage centrifugal fan, driven by a $\frac{3}{4}$ hp input, 16000 rpm universal motor. With a closed orifice, water lift is 61 inches. Vacuum cleaner is made by Lewyt Corporation, Brooklyn, N. Y.

VACUUM CLEANER and attachments fit in a specially designed storage container. Canister unit, wands and hose fit in lower compartment. Attachments are carried in a separate top compartment.



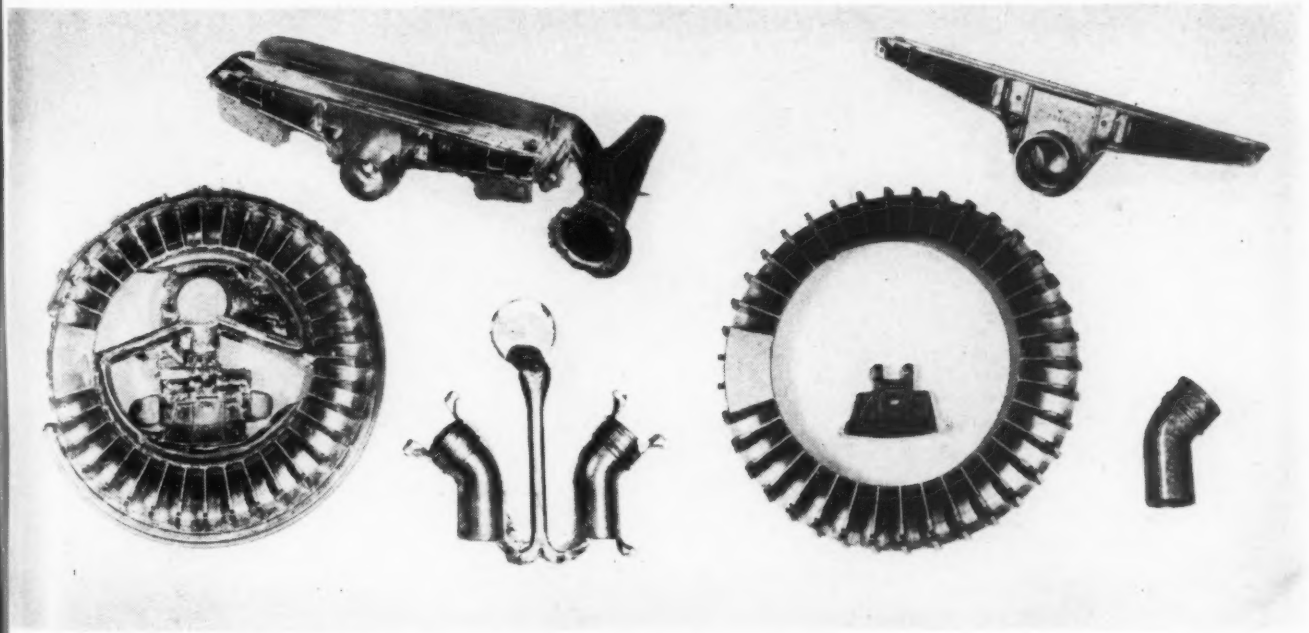
CANISTER UNIT CROSS-SECTION shows the vertical mounting for the motor. Since the cleaner is designed to operate with 2 or 3 inches of water in the bowl, cleaner components are subject to moisture conditions. For this reason, lower cloth filter support is a rubber molding which, of course, does not corrode as would a steel frame. Fins are die cast in motor bearing housings to promote bearing cooling process. Air flow from the fan is divided; some passing through the motor, and the rest through openings in the unit that direct it around the motor and into the top compartment. Motor unit is made by the Lamb Electric Company, Kent, Ohio.



STEEL DUST BOWL is cut away to show the filter system. Three filters are provided to remove dirt particles from the air as it flows through the unit. Tabs on fiber filter are crimped in bumper ring when top is clamped on bowl.

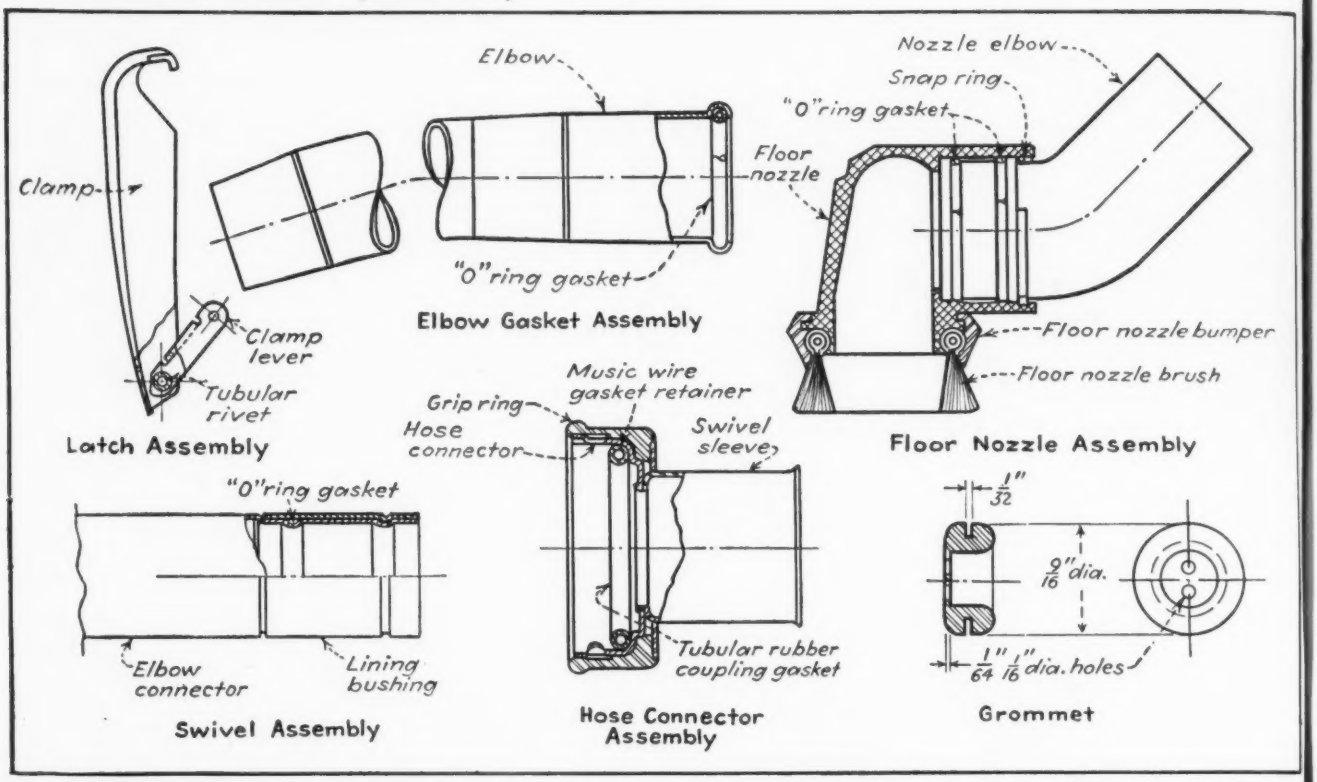
OPERATING DESCRIPTION OF AIR FILTER SYSTEM

Air enters the bayonet lock adapter fitting and is deflected by an aluminum baffle plate, which imparts a centrifugal motion to air entering the unit. In addition to inducing this centrifugal air flow, baffle prevents solid objects from travelling into the filter and piercing it. First filter is a cone-shaped chemically treated paper made of rayon and hemp, which is replaced after long periods of service. This filter removes the majority of the dust particles from the air. A chemically treated cloth filter further removes dirt from the air. This cloth filter is also cone-shaped and is stretched over a frame, thus acting as support for the paper filter. The air flow from the cloth filter is directed to cool the motor. From the motor the air travels through a third filter made of treated cellulose fibre. This third filter acts as silencing material and also serves to remove from the air, carbon particles that constantly wear from the motor brushes.

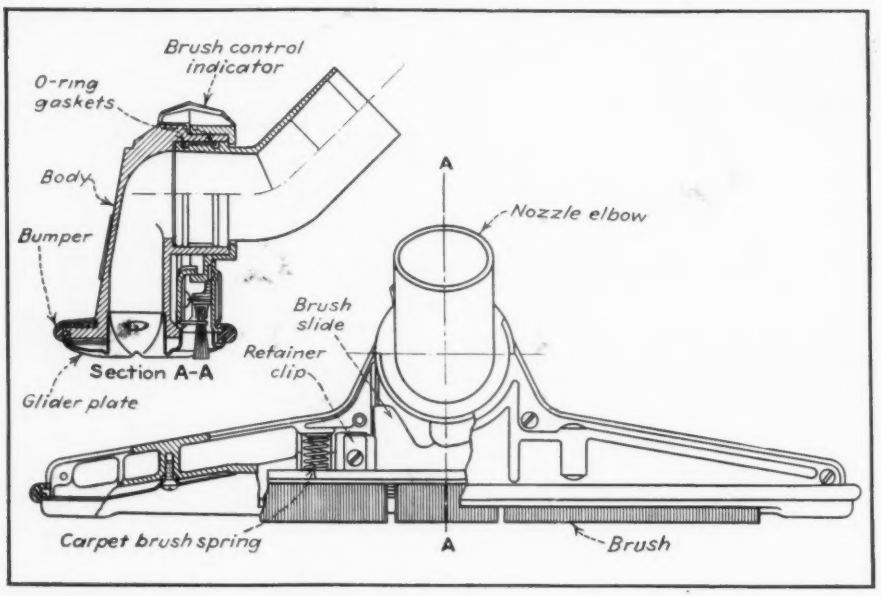


ALUMINUM DIE CASTINGS are used in the majority of the cleaner components. Aluminum die castings were selected because of their light weight and the relatively few machining operations required by die cast parts. Cored holes are cast in the parts for self tapping screws. Some of these die castings are illustrated before and after trimming. These illustrations show how parts can be grouped in a die to reduce production cost. No machining is done on these parts after the trimming press operation. Fin ring and switch pedal are cast in the same die. Two nozzle elbows are also cast together. Outside edges of fin ring sections are machine polished.

Vacuum Cleaner (continued)

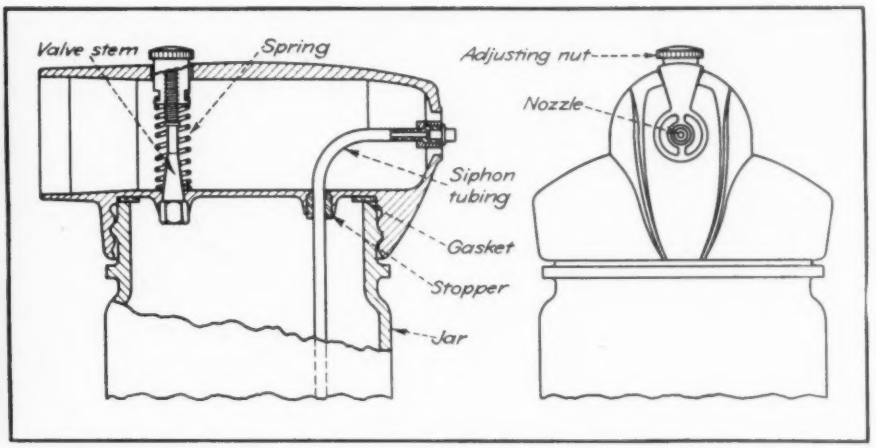


DESIGN FEATURES include the use of O-ring seals made from vinyl plastic tubing with the ends heat sealed together. Hose connection assembly has a tubular rubber seal held together by a music wire retainer inside the tubing. A similar design is used for the cover seal. To avoid air escape around lead wires from the motor, a special grommet was designed from natural rubber. This grommet has a conventional ring cross section with a diaphragm section closing the ring. Wires are drawn through two holes in this diaphragm that are smaller in diameter than the lead wires. An added design feature is the latch assembly for clamping the motor unit on the bowl. This latch is designed so that as the latch comes backward from the bumper pad, it does not touch highly finished surfaces of the unit.



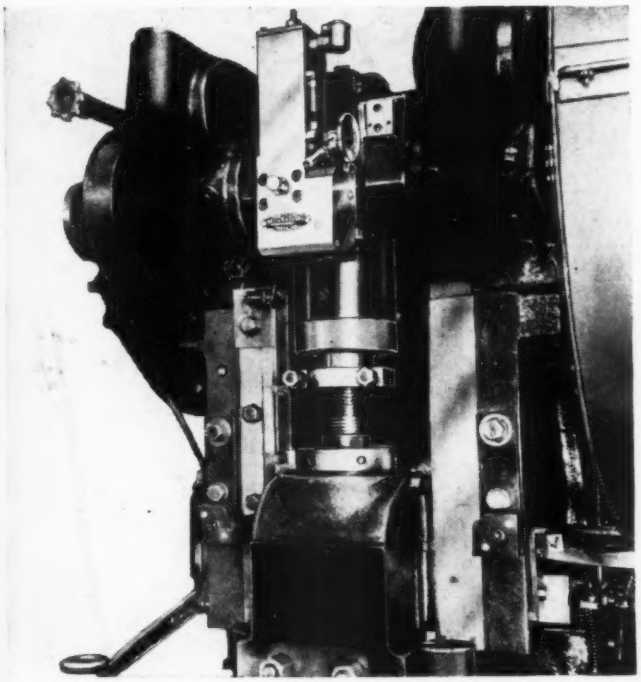
CARPET NOZZLE ASSEMBLY is constructed from materials that resist corrosion. Bumper is vinylite plastic, selected to avoid leaving black marks on furniture. Rug glider plate is stainless steel. Fuller brush pivots in operation and is spring loaded with beryllium copper springs. Phospher bronze clips are used. Two O-ring gaskets are also used in this assembly.

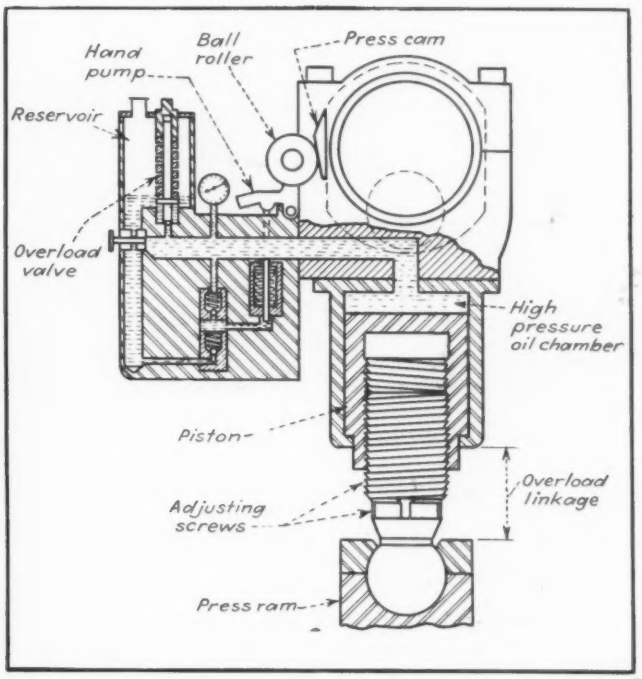
HANDLE has cast cooling fins. Depth control dial is die cast zinc, chosen to reduce sliding friction with aluminum nozzle body. Recesses act as popper stops for adjusting brush height.



PAINT SPRAYER HEAD is a zinc alloy die casting. Threads are cast in the piece. Vinyl tubing that extends to the bottom of the jar permits all paint in the jar to be applied. Sprayer operates by admitting air into the jar through an adjustable brass cone-seated valve. Pressure on top of the paint in the jar in addition to the reduced pressure at the end of the nozzle causes paint to flow from the tubing into the air stream at the nozzle.

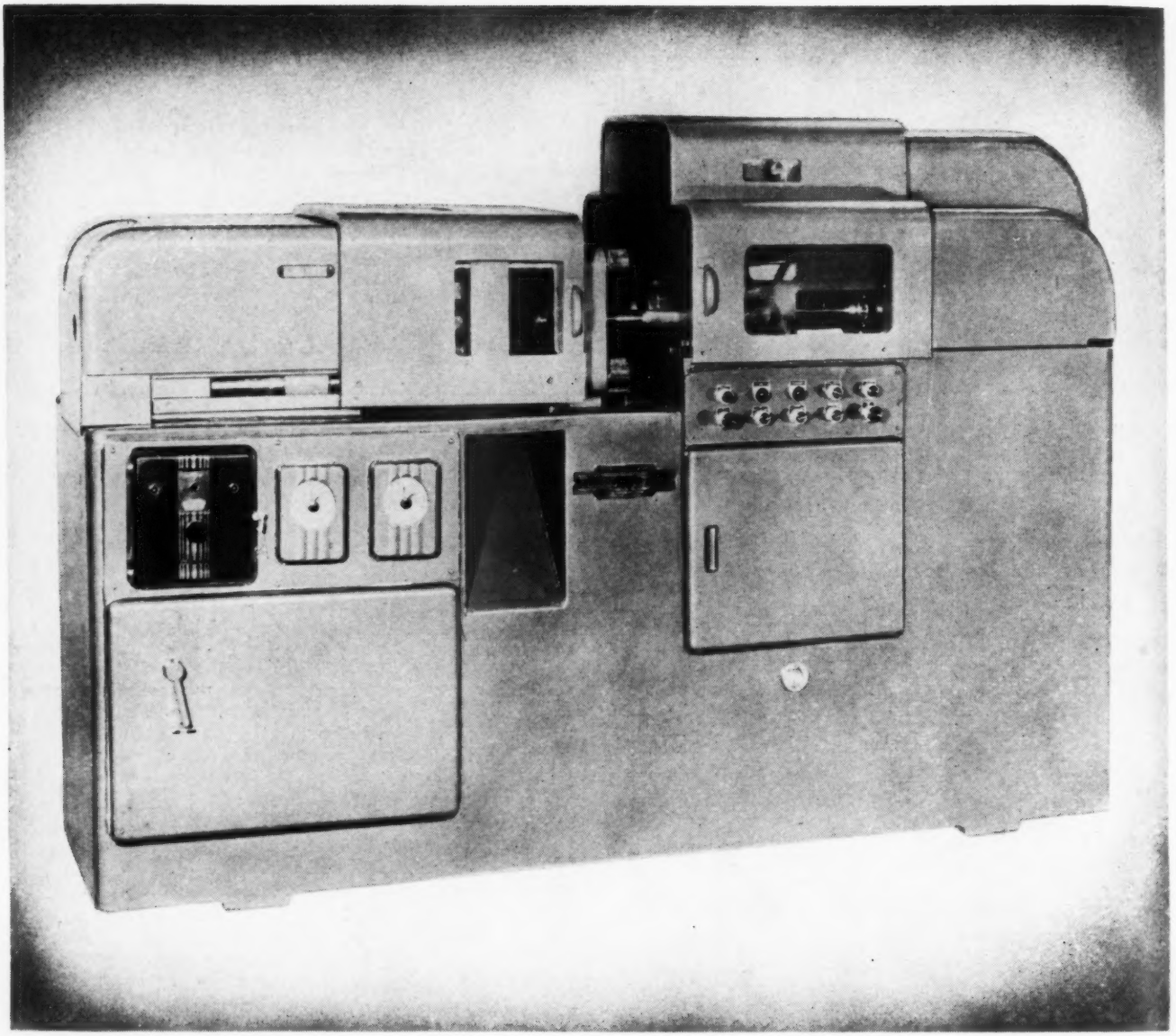
Telescoping Hydraulic Connecting Rod





BY REPLACING the connecting rod of standard punch presses with a telescoping hydraulic connecting rod assembly, damage to press cranks and frames resulting from overload can be avoided. The hydraulic overload jack, made by Dayton Rogers Mfg. Co., Minneapolis, Minn., replaces standard crank and adjusting screw assemblies. This unit is the connecting link between crank bearing and ram and is calibrated at the maximum tonnage of the press. It can also be calibrated to protect a group of tools used in the press. The overload jack can be installed on single and double crank punch presses. It is built in sizes from 25 to 500 tons working capacity.

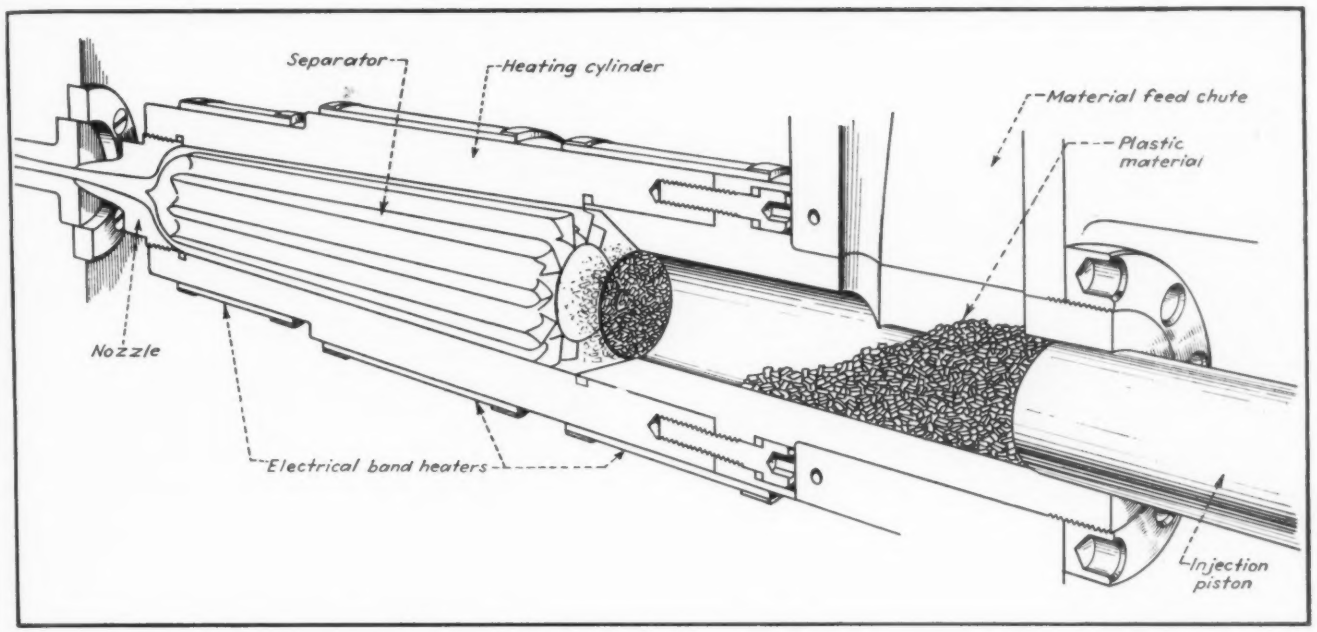
HYDRAULIC JACK telescopes when overload occurs. If the preset working pressure of the press ram is exceeded during the working cycle, the piston moves in the high pressure oil chamber, permitting the rod to shorten in length. Oil escapes through the overload valve to the reservoir. Hand pump is actuated to reestablish high pressure in the chamber. Ball roller is actuated from the press cam and maintains constant pressure in the chamber. Adjusting screws are provided for setup adjustments. A hydraulic limit switch that will cut off the motor drive for the press when overload occurs can be installed in this unit.



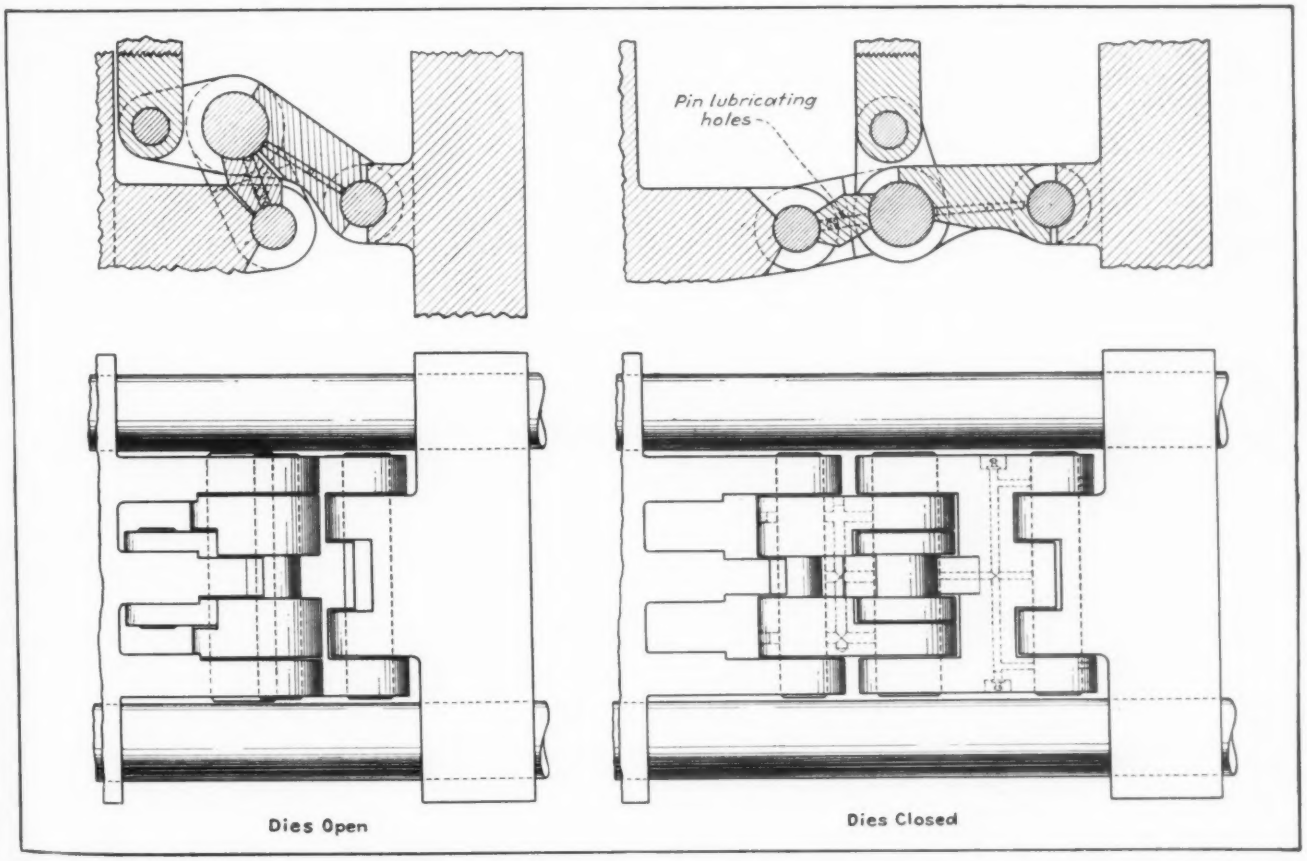
Heating Cylinder Has Tapered Bore Design

A tapered bore heating cylinder in the Model 1B-2 Fellows-Leominster Injection Molding Machine improves heat distribution and reduces the possibility of burning the plastic material. Designed for production runs of small industrial hardware, jewelry and novelty items or for short runs of larger parts, the machine is fully automatic. Two oz of plastic material can be molded per shot and the machine makes six shots per minute. Injection pressure on the material is up to 20,000 psi. Hopper capacity is 25 pounds. Toggle mechanism gives a locking force of 150 tons. A vibration proof temperature control instrument is built into the machine.

The injection molding machine is built by the Fellows Gear Shaper Co., Springfield, Vt. It is approximately 8 ft long and weighs 7,800 lb. A standard Vickers hydraulic system has controls for speed and pressure of injection. Heating cylinder, hydraulic pressure cylinder, hopper and feed mechanism are advanced and withdrawn hydraulically as a unit. This type of construction facilitates setup and operation by permitting the operator to withdraw the nozzle for purging the heating cylinder or removing a wedged sprue. Nozzle slide is held in place when molding by a locking mechanism. Volumetric control of material feed to the heating cylinder has a thumb nut for adjusting measuring slide. Heater units have a maximum power consumption of 2½ kilowatts.



"SPEED FLO" CYLINDER has a tapered bore and separator. The larger mass of the separator at the feed end stores heat for plasticising the material where it enters in the cold state. Reduction in the cross section of the flutes toward the small end of the separator is proportional to the reduction in volume of material. The heating units are arranged to fit this condition. A subsequent reduction in heat voltage input of 15 percent is claimed for this type of design. Another advantage of the tapered separator is its close fit to the cylinder bore, which prevents plastic material from lodging between the separator and bore. If plastic material lodges it burns and discolors.

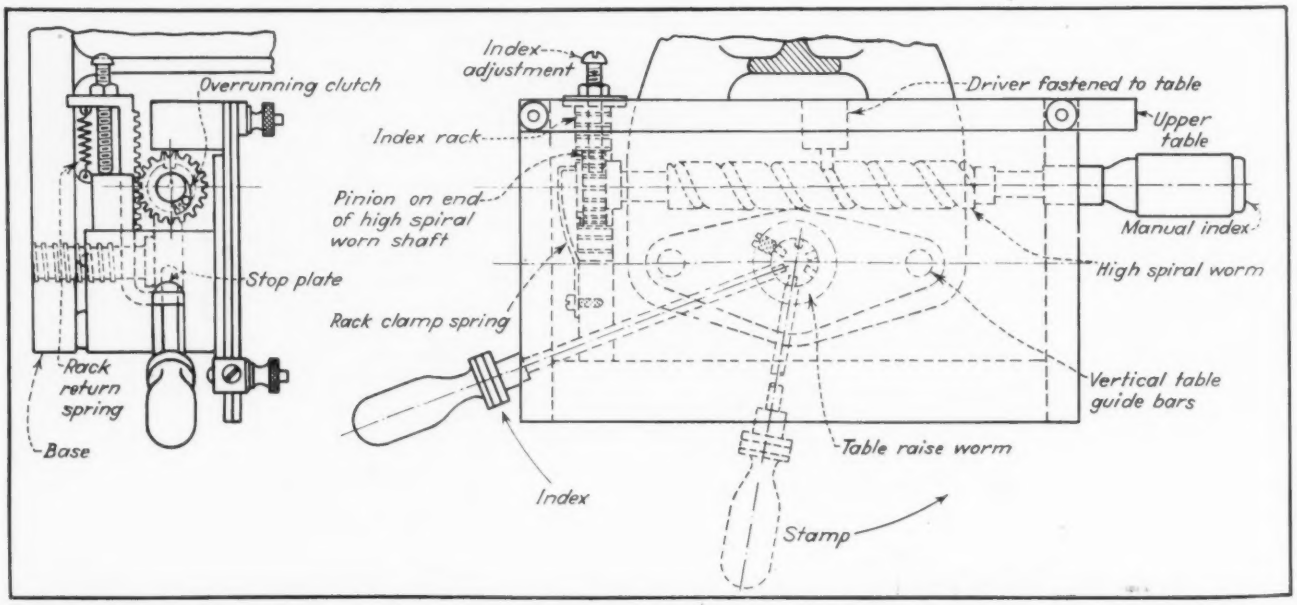


TOGGLE MECHANISM is designed with full support for the length of the link pins. This type of design avoids pin breakage and assures that the dies are positioned firmly in contact. Links are drilled to lubricate pins.

Stamping Machine Has Simple Table Movements

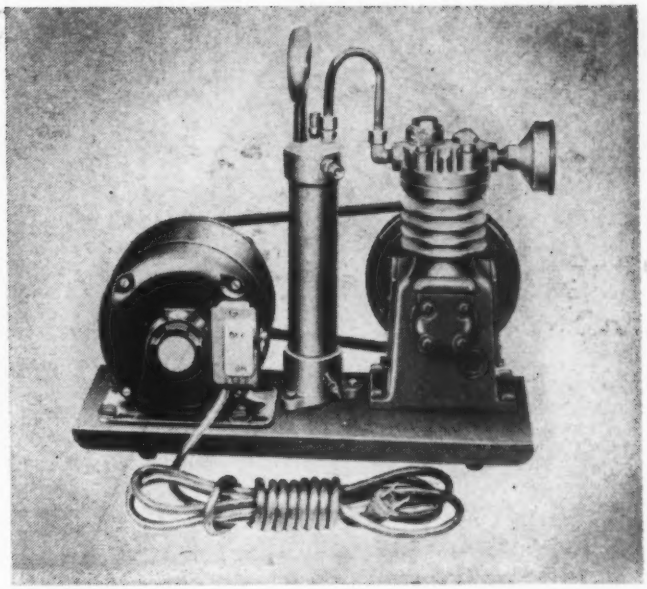
A single lever on this stamping machine raises the table for the stamping operation and indexes the table for the next stamping position. Known as the Acromarker Model No. 1, this machine is designed to stamp nameplates, scales, dials and gauges. A worm driven circular index table unit can be supplied for stamping numbers in a circular path. Thickness adjustment up to ⅝ in. is provided. In larger models adjustment up to 2½ in. is obtained. Table has 3/32 in. lift to stamping dies. Machine is made by The Acromark Co., Elizabeth, N. J.

STAMPING MACHINE is operated by moving the left hand lever to the left for table index and to the right for the stamping operation. Right hand lever is turned to manually index the table. Cast iron index plate is mounted in double row precision ball bearings in a cast iron frame. Stamps are clamped in radial grooves by a bronze faceplate. Plate is manually indexed to spring loaded poppet stops. Vertical travel adjustment is made by fitting handle in different holes in ground steel worm shank.



HAND INDEX OF TABLE is made by turning the right hand handle, which turns a high spiral screw that engages a pin mounted on the table. Automatic index is accomplished by first disengaging rack clamp spring, which permits rack to engage pinion on end of high spiral worm shaft. Table index is then made by moving the handle to the left. When handle is moved to the right for the stamping operation, an overrunning clutch in the pinion permits the spring loaded rack to return to position for the next index without turning the worm.

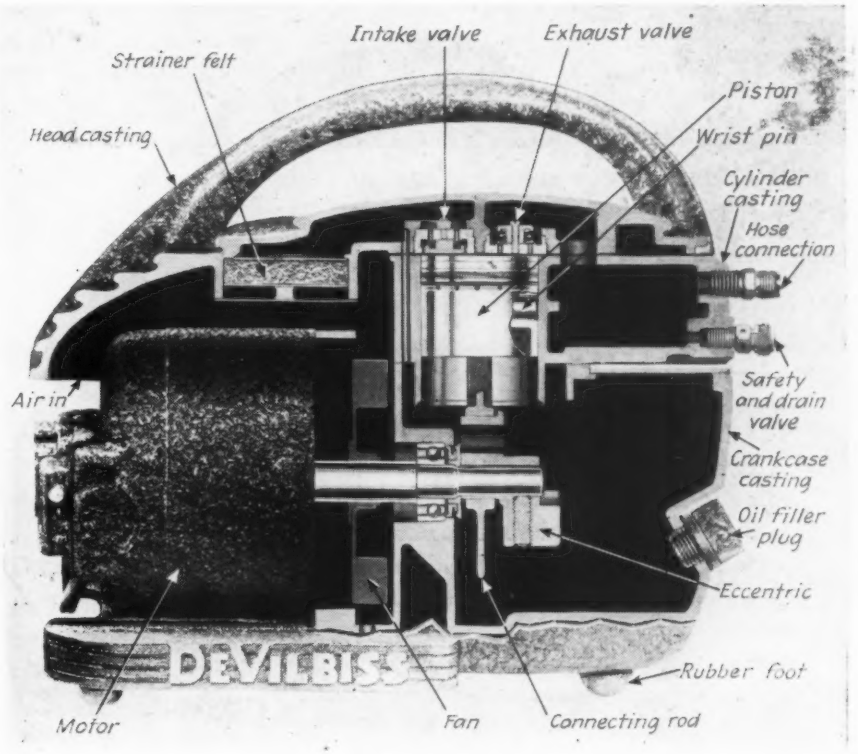
NEW UNIT is single cylinder ¼ hp air compressor that has handle cast in cylinder head. Air delivery with a-c or d-c motor is 2.6 cfm at 40 psi pressure. Unit is finished in two tone brown and orange wrinkle paint. It is 15 in. long by 6 5/16 wide by 12¼ in. high. Splash lubrication is used.



FORMER UNIT had a V-belt driven single cylinder compressor. Air intake assembly was mounted on the cylinder head. Handle was mounted on pulsation chamber that was included to maintain constant pressure in the line. Unit was 18 in. long by 11⅜ in. wide by 15⅛ in. high.

Three Piece Cast Iron Compressor Unit

This portable air compressor unit for paint spraying operations was made by redesigning a conventional motor driven unit mounted on a flat base. The conventional unit was neither compact nor inexpensive to manufacture. By inclosing the entire unit with a three piece cast iron housing, appearance and utility were improved in addition to reducing production cost. One cylinder and two cylinder units of this type are manufactured by The De-Vilbiss Co., Toledo, Ohio. Styling was by Sundberg-Ferar, Detroit, Mich.



NEW UNIT has compressor direct driven by the motor. A motor end shield is cast in the crankcase and the motor flange mounted on a pilot on the end shield. Ball bearing that supports compressor end of motor is mounted in the crankcase casting. Air filter, intake and exhaust valves and handle are included in the cylinder head casting. Cooling fins are provided in both cylinder and cylinder head castings. Pulsator chamber is included in the cylinder casting. Safety and drain valve is mounted in the cylinder casting.

NEW MODEL 300 Cummins Perforator has a cast aluminum housing and frame. It weighs 44 pounds. Production time to make this model is 33⅓ percent of that of former model. Material costs are 21.9 percent of those of former model. Machine is actuated by insertion of paper in die block opening or by manual button control.

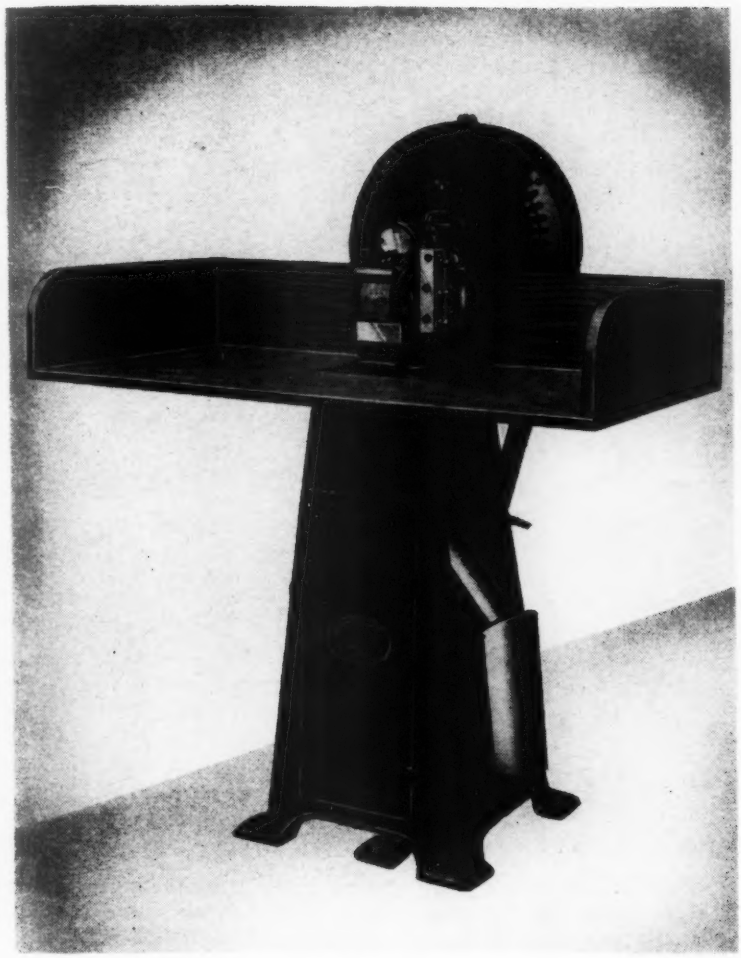
Perforating Machine Components Redesigned

The Cummins Model 300 Perforator is an example of a product completely redesigned to reduce product cost and improve utility and appearance. A portable model has replaced a former floor-type model and performs the same operations of perforating checks, bills, documents and stamps for cancellation. A new driving mechanism that operates only when the punching operation is performed, has been designed. Styling was by Raymond Loewy Associates. The machine is manufactured by the Cummins Business Machines Corporation, Chicago, Illinois.

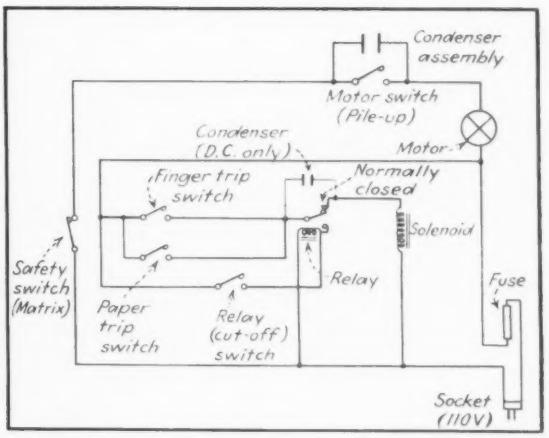
Punches in new model are depressed by an eccentric shaft mechanism that is worm gear driven by a ⅜ hp 11,000 rpm series stop and start motor. Marking characters are selected by depressing matrix keys, which permit the circular sintered metal matrix rings to rotate. A safety switch is provided to prevent the machine from operating if character selected is not properly aligned. Waste drawer is forced out of its guides and opens if the drawer is not emptied. The perforator is 8 in. wide, 13 in. deep and 11 in. high, and finished with wrinkle gray enamel.

Former model had a punch press type flywheel mechanism, which was driven by a ¼ hp, 1140 rpm motor that ran continuously. Matrix rings were made from solid bars. They were set by removable pins which, if lost, would permit the selection of improper punch combinations. Punch waste was caught in a bucket which, if not emptied, permitted the waste chute to fill with waste, resulting in serious damage to the machine. It had a smooth baked black Japan finish. The cast iron frame required added finishing operations to apply this finish. The machine was 19 in. wide, 20 in. deep and 53½ in. high.

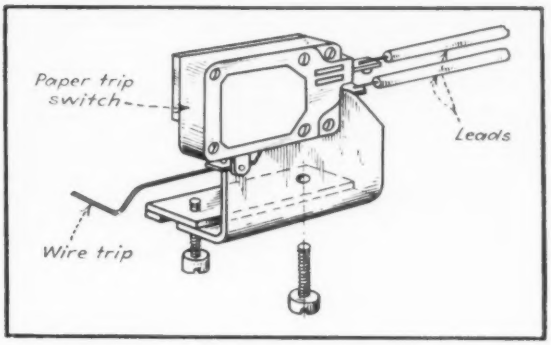
D E S I G N S



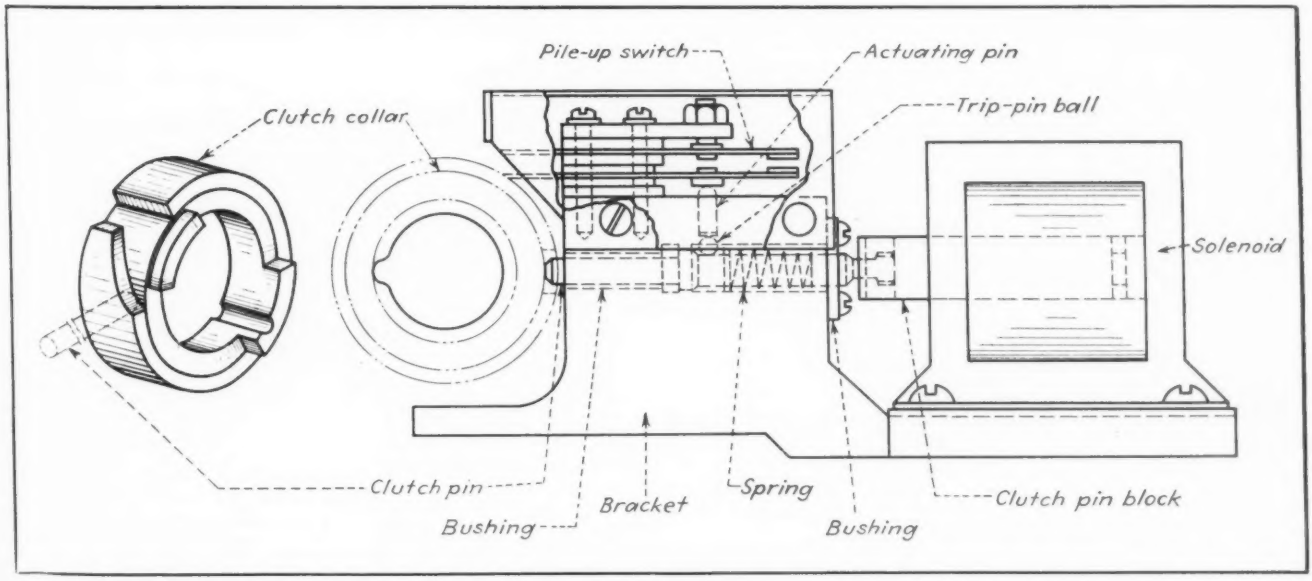
FORMER MODEL 75 weighed 512 pounds. It was pedal operated and had a punch press type drive with a motor that ran continuously. List price was approximately 1½ times that of new model machine.



WIRING DIAGRAM shows the relay cutoff switch that is closed by the linkage movement, thus moving the relay from its normally closed position and de-energizing the solenoid.

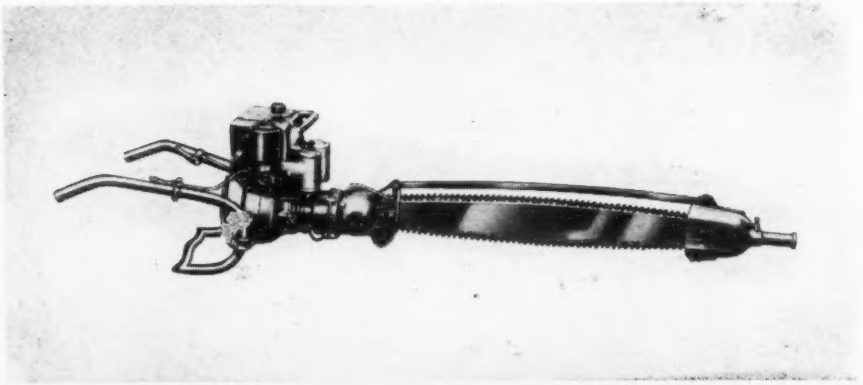


PAPER TRIP SWITCH operates when a check is inserted in die throat. Paper lifts wire trip and operates switch to start operating cycle.

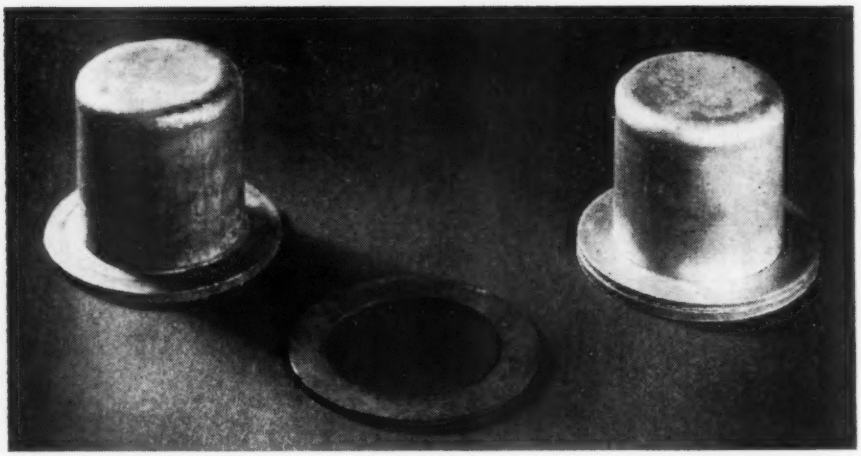


ELECTRIC SWITCH has solenoid that withdraws clutch pin from collar. This releases collar, permitting it to engage teeth on moving member. Clutch pin forces up actuating pin and operates pile-up switch to start motor. Relay opens after cycle starts and de-energizes solenoid. When clutch pin drops in cam slot in collar it disengages clutch and shuts off motor. Cycle cannot be started again until solenoid is re-energized by operating a trip switch.

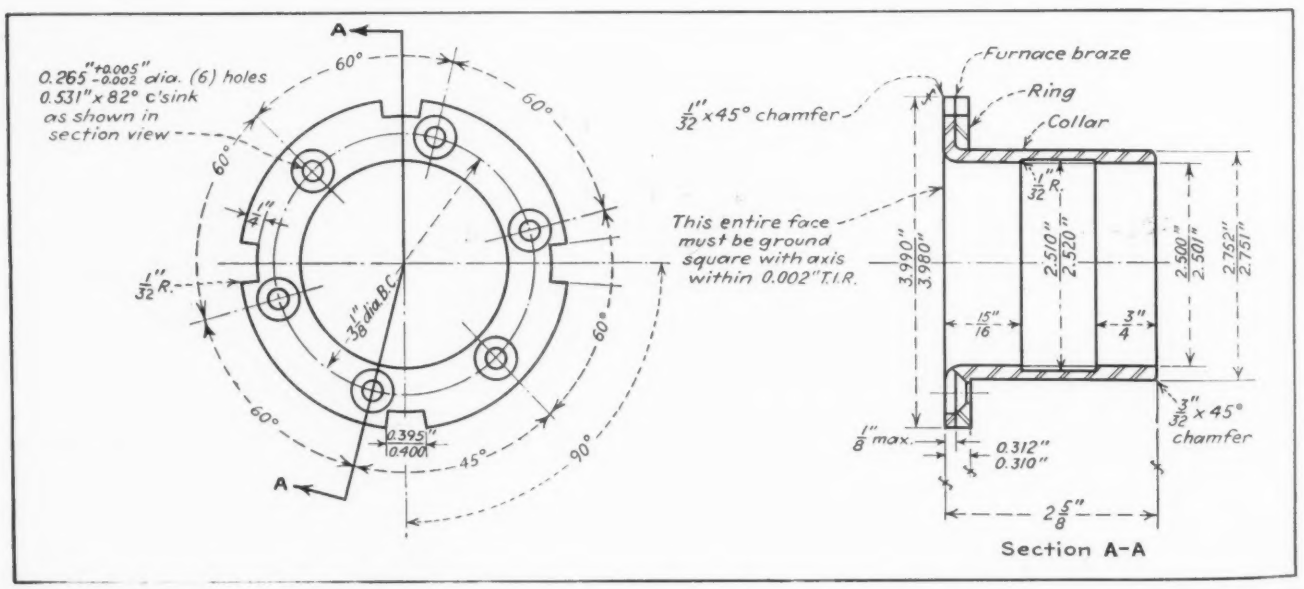
Chain Saw Index Sleeve Redesign Reduces Cost



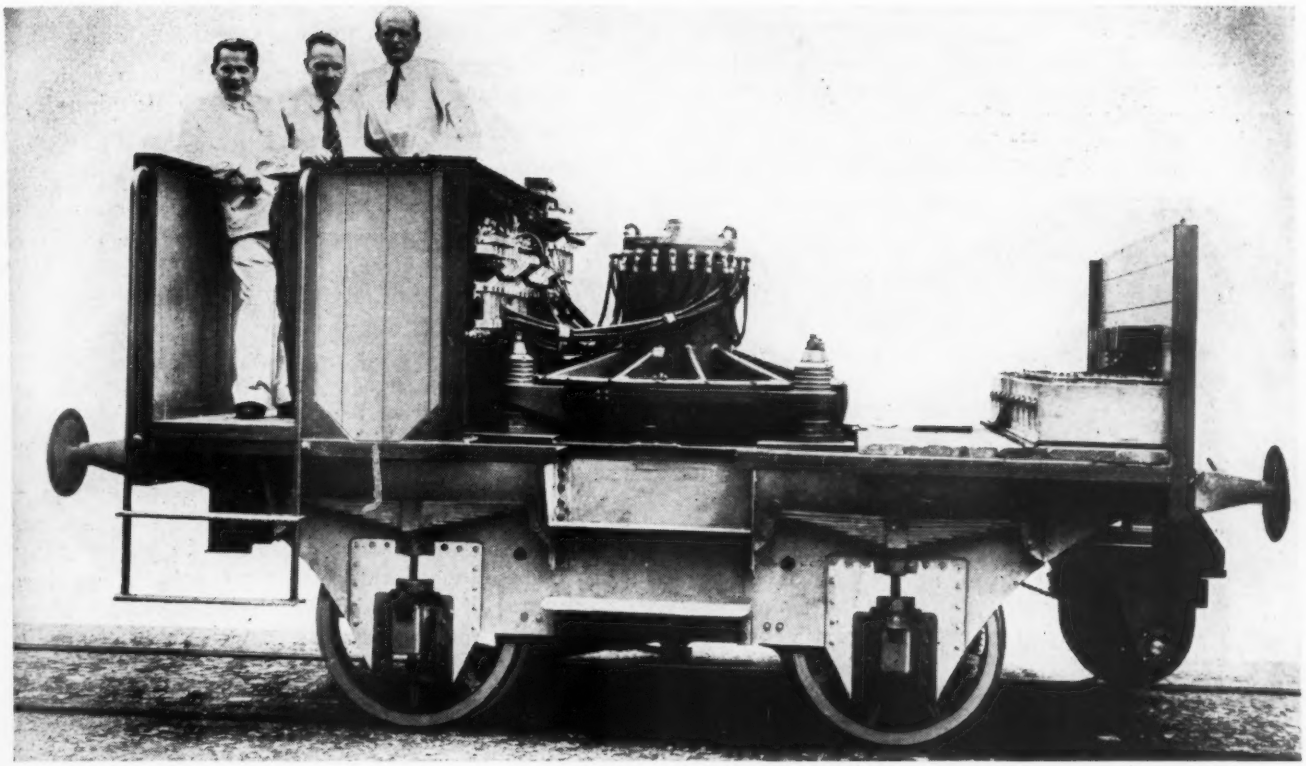
Light-weight of this 76 lb, 24 in. power chain saw was achieved by specifying stampings and light-weight metals. Production cost of the index sleeve, about which the chain swivels to four positions, was reduced by specifying a brazed stamping assembly. Magnesium alloy crankcase and aluminum alloy piston and guard bar are provided. Saws are made in capacities from 18 in. to 12 ft by the Mall Tool Co., Chicago, Illinois.



SLEEVE COMPONENTS include a cup stamping and a reinforcing ring that is copper hydrogen brazed to the index sleeve flange. Index sleeve is made from hot rolled strip steel stock, dead soft temper hardness, 54 Rockwell "B" scale or under, 0.187 in. thick. Ring is made from hot rolled strip steel stock, half hard temper, 72-80 Rockwell "B" scale.



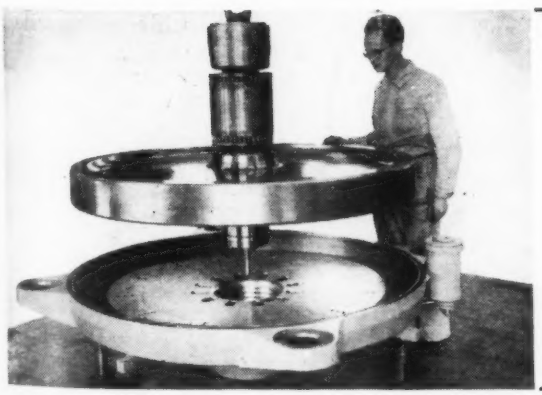
INDEX SLEEVE was originally machined out of solid stock. It was redesigned first to be made from tubing stock as a stamping. This redesign reduced the cost of the part below that of the former design. A second redesign was made by specifying a cup stamping with a stamped reinforcing ring brazed to the flange. After brazing, the bottom of the cup was cut off and finish machining operations performed. The resultant savings per piece were approximately 70 percent of the cost of the original design.

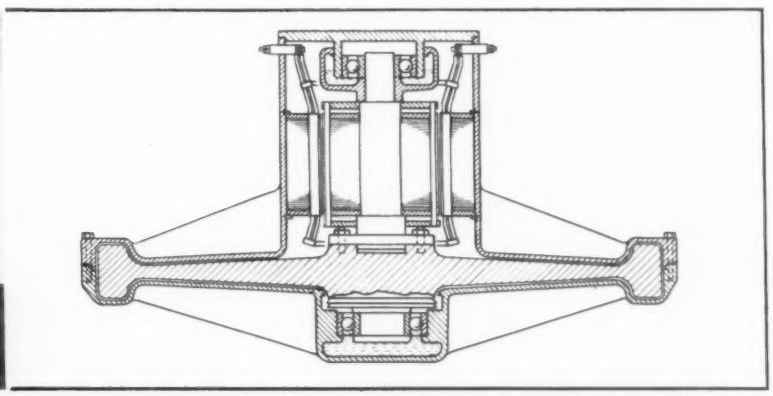


ELECTROGYRO vehicle energy unit has the rotor of a pole-switchable, d-c, short circuit-anker motor coupled to a flywheel. Flywheel is brought up to 3000 rpm•speed by connecting to a normal low tension direct current source. During runs, the motor is excited by condensers and delivers, as a generator, direct current to the driving motors.

Flywheel Energy Source for Electric Vehicle

By using stored energy in a flywheel to supply electric current for a motor driven rail car, storage batteries or overhead lines are avoided. An experimental electrogyro unit employing such a principle has been developed by Ateliers de Construction Oerlikon, Zurich. Tests have shown that a 10 ton vehicle having a 1 ton flywheel can be driven 6 to 10 miles on one charge of the flywheel. Charging process takes 1 minute.





FLYWHEEL is made of chronium-nickel steel. It is mounted on ball bearings inside of a hydrogen filled casing to reduce friction losses. A period of from 10 to 15 hours is required for the flywheel to run out when the vehicle is idle. A suggested practical use for such a driving unit is for mine work where overhead lines are dangerous and also difficult to construct. For passenger vehicle service, there can be no grades and frequent regular stops must be provided, as in suburban street car or bus lines, for recharging the flywheel.

PROGRESS IN REVIEW

In this running summary of technical developments, space restrictions do not allow giving complete details. Further information relating to any item can be obtained by writing to the Editor of Product Engineering.

PLASTICS AND RUBBER

MATERIALS SHORTAGES continue to affect the plastics industry. Ironically, it was the short supply of metals and other scarce materials during the war that led to the increasing use of plastics as substitutes and supplements. Now that plastics are established as basic engineering materials in their own right, scarcity of raw chemicals that go into plastics hamper production. Shortages include not only raw materials to synthesize the basic resins, but also vital supplementary ingredients, such as plasticizers and fillers.

■ Phenolic resins have long been in short supply because of the scarcity of phenol and formaldehyde—it now appears that melamine resins may become tight because of insufficient raw melamine; even urea resins may be affected.

■ The filler problem may find its answer in the products of agriculture. A new molding compound, Plastone A, developed in the Cotton Oil Laboratory of the University of Tennessee, utilizes cottonseed hulls as a filler. Still another molding compound contains pecan shell flour.

■ Natural resins may be another answer to the shortage of synthetics. Use of rosin has been expanding rapidly. Made from pine stumps by the Crosley Chemical Plant in Louisiana, it is thought that there are sufficient supplies in this one locality for another thirty years of rosin production.

IMPROVED PRODUCTS and processes are on the increase, regardless of raw materials shortages. As in other fields, research is continuous, and because of the newness of the industry, old-fashioned ideas are rapidly discarded and brand new research techniques are unhesitatingly applied.

■ Although plastics technology is relatively new, a formidable body of literature in this field has already appeared. To cover the study of plastics by means available in modern science, T. G. Rochow, of the American Cyanamid Company, proposes the term "resinography". This might include the fields of chemistry, physics, botany, metallurgy and minerology.

■ Electron microscope techniques are now used in plastics research. These methods permit detection of surface "discontinuities" of less than 2×10^{-8} in. Important results will be seen in plastics products requiring smooth surfaces.

STRUCTURE OF HIGH POLYMERS is being studied intensively at various industrial and university laboratories. While plastics technology has not yet "jelled", as for instance in metallurgical studies, certain broad generalizations have been recognized.

■ Professor H. Mark of Brooklyn Polytechnic Institute puts all polymers in one of three classes: (1) "Fiber-formers", polymers having a softening point above 480 F, a strong tendency to form crystals which can be readily oriented; (2) rubberlike or elastomeric polymers, having a softening point below —40 F, with a tendency to crystallize when stretched (not crystallizing if unstretched); (3) the "usual" plastic type of polymer, having a softening point from 200 to 400 F— showing no crystallization under any conditions.

■ Polymerization of vinyl plastics is explained in the December issue of *Chemical Engineering*, in terms understandable to any engineer. Main methods of polymerization include the mass, solution, emulsion, and suspension ("pearl") techniques. Effects of temperature control, catalysts, emulsifiers, and modifying agents are discussed.

■ Behavior of rubber-like substances is explained by the Japanese with the aid of statistical mechanics in the *Journal of Colloid Science*. This theory predicts a relationship between double refraction and elongation which coincides exactly with that obtained experimentally for linear macromolecules such as polyisoprene, polyvinyl chloride and polystyrene. The predicted relation between stress and elongation also fits actual data rather well. However, other sources report that some elastomers depart considerably from predictions of the statistical theory of elasticity. For example, some torsion experiments have indicated that stress is proportional to absolute temperature.

PRODUCTION TECHNIQUES continue to be improved. Vital factors that originally hampered progress in the plastics industry were haphazard molding methods and lack of standardization. This was true not only of fabricating conditions, such as temperatures, pressures and mold design, but applied also to the machines themselves. Testing methods also needed improvement, and such organizations as the Plastics Materials Manufacturers Association and ASTM's Committee D-20 have devoted research to this end.

■ A comprehensive plastics research program at M.I.T., sponsored by the Plastics Materials Manufacturers Association, is already bearing fruit. A new universal testing machine has been developed especially for plastics. Through servo-mechanism controls operating via electrical, mechanical and hydraulic circuits to move a double-loaded hydraulic piston, load against rate of strain, or extension vs rate of load application can be recorded.

■ Special features of the new plastics testing machine include: (1) Continuous operation from compression to tension without resetting the grips or extensometer

—thus recording a complete hysteresis picture; (2) the load weighing mechanism has extremely low inertia; (3) recording is done accurately over a wide range; (4) the recording system is stable over a long time interval, allowing accurate creep and relaxation data to be obtained; (5) the machine is satisfactory for measuring tensile, compressive, bending, shear strengths, some types of fatigue and high velocity creep or low velocity impact.

■ Although the general advancement of the plastics molding art is recognized, we do not often realize that plastics is a relatively new field in foreign countries. The first injection molding machine was recently installed in the Philippine Islands, despite difficulties of fluctuating voltage and lack of replacement parts. This particular machine will be used only for polystyrene, since other thermoplstics are claimed to have inferior dimensional stability in the tropics.

ELASTIC COMPOSITIONS can be molded just as readily as hard plastics. This is perhaps a clue to the "overlap" between rubber and plastics. New compositions, such as the silicones, have also contributed to the common ground between the two groups.

■ Silicone rubber has been molded into gaskets, used between supercharger connections and each of 16 cylinders on the new General Electric-American Locomotive Company diesel locomotive.

■ Latest outlet for silicone gaskets is the O-ring gasket which provides an effective oil and air vapor seal on the TG-180 jet engine. Silicone gaskets, in addition to excellent thermal and electrical resistance properties, will maintain original characteristics even after long periods under vacuum and have no tendency to adhere to metal or non-metallic surfaces. They can, however, be bonded with special adhesives.

■ Silicone "bouncing putty", a laboratory curiosity for several years, has finally been put into commercial use. Its property of bouncing upon impact yet flowing under constant pressure is the basis for an automatic leveling device for furniture. The putty is contained in simple piston and cylinder arrangements known as "Levelmatics," that automatically adjust table leg lengths to uneven floors or rugs to the extent of 3/16 in.

PRODUCTION of natural and synthetic rubber continues at a record-breaking volume. Associated with this high production are signs of health in the rubber industry. New and better synthetic rubbers are being developed.

■ Demand for rubber products will continue at present high levels during 1948, according to a statement of the United States Rubber Company. Total rubber industry output is expected to exceed 2½ billion dollars this year—more than 2½ times the 1940 volume—although slightly less than the all-time high of 2.7 billion dollars reached in 1947.

■ There are now over 400 experimental variations of GR-S rubber in regular production by agents for the Office of Rubber Reserve. R. H. Harmon of the O.R.R. in an article in *Rubber Age* explains why there are so many types. Although most GR-S types are styrene-butadiene copolymers, a few contain only one or neither of these two raw materials. As for labels: XP prefix means produced in laboratory pilot plant, X prefix means large scale production, while regular GR-S numbers without prefix letters denote a material already produced to the extent of a million pounds and still in demand.

■ Recent Congressional hearings aired government claims that private industry in the U. S. would not be able to produce GR-S below the present 18½ cents per lb government-fixed price. An R.F.C. administrator stated that 14½ cents was needed to cover out-of-pocket expenses for materials and labor and that the other 4 cents per lb was based on depreciation and other charges. According to his report, the U. S. government to date has suffered a loss of nearly 250 million dollars in its synthetic rubber program, directly chargable to depreciation of plants.

■ Two new synthetic rubbers produced by private industry are "Philprene A" and "Philprene B." The former is claimed to be superior to both natural and other synthetic rubbers for tire treads while the latter is designed as a replacement for natural rubber in the carcasses. They are synthesized from the same raw materials as regular GR-S, by what is claimed to be an improved process. Produced only in pilot plant to date, field trials reveal superior wear and crack-resistance for A, and superior tensile strength and crack resistance for B.

NONFERROUS METALS

ALUMINUM, most abundant of nonferrous metals, is being applied more than all others. In 1947, estimated production of primary aluminum topped all previous peacetime records with 1.14 billion lb. Knowledge of aluminum uses is piling up almost too fast for thorough sifting and evaluation. Alloy contents, tempers, methods for testing and analyzing, and pure metal uses are still in a state of flux.

■ Large tonnages of primary aluminum were used to replace copper wiring. High tension transmission wires, with higher strength, were found to be excellent—especially in mountain regions.

ALUMINUM ALLOY SELECTION has become a difficult problem for designers. With others, Herbert Dobkin, writing in *Steel*, has tried to classify aluminum base alloys. The task becomes more Herculean each day.

■ Aluminum-manganese alloys, such as 3S, do not respond to heat-treatment and should not be used where high strength is necessary. However, they are excellent for piping and kitchen utensils.

■ Aluminum-magnesium alloys, such as 52S, that do heat, treat, have strengths between commercially pure and high strength aluminum alloys.

■ Aluminum-copper-magnesium-zinc alloy 75S is highest strength aluminum base alloy when properly heat-treated. However, it must be cladded for good

corrosion resistance, good heat-treating characteristics and high strength. Readily worked and suitable for forgings.

■ Dairy and brewery fittings are made from heat-treatable 51S and 53S corrosion resisting aluminum-magnesium alloys.

■ Alloy 18S, aluminum-copper-magnesium-nickel, can be used for pistons and other parts up to 400 F. Physical characteristics similar to 25S alloy.

ELEVATED TEMPERATURE STUDIES into physical properties of 24ST and 75 ST have been carried out since the latter is replacing the former in the aircraft field. Separate investigations are reported by M. L. Ochieano in *Machinery,* and J. A. Nock, Jr., in *Iron Age.*

■ Cold 75ST not as formable as cold 24ST, but upon heating for ½ hour at 300 F, formability equal to cold 24ST is obtained. No loss in corrosion resistance or strength after this heating.

■ Following effects noted for 75ST and Alclad 75ST sheet aged 24 hours at 250 F and then reheated 1,500 hr: 225 F, no loss in strength; 250 F, 5-10 percent loss; 275 F, 20-25 percent loss, and 300 F, 30 percent loss. Such reheating had no marked effect on corrosion resistance or resistance to stress corrosion cracking.

■ 24ST and Alclad 24ST36 sheet were heated at 250 F for 1,500 hours with no loss in strength. At higher temperatures strength did fall.

PLATING OF ALUMINUM, is seldom done to increase corrosion resistance of the part. A staff article in *Metal Industry* indicates plates are used on aluminum to: Increase eye appeal, improve electrical contacts or permit rubber bonding.

■ Some plates are applied directly over anodic coatings. This can be done with nickel, copper, cadmium, silver and iron, if anodized in phosphoric acid electrolyte under carefully controlled conditions.

■ High purity aluminum alloys, die-castings and sand-castings must first be coated with zinc by immersion in a solution of sodium zincate and caustic soda, then given a copper strike applied by Rochelle salt type solution, and finally plated with nickel, cadmium, zinc, brass, silicon or chromium.

ELECTROPLATING TECHNIQUES are now aimed predominantly at utilitarian rather than decorative applications, according to Adolph Bregman, writing in *Iron Age.* There are six outstanding examples of this trend.

■ Hard, porous chromium plate used in cylinder walls of diesel engines retain lubricating medium and reduce contact area between sliding parts.

■ Simplification of engineering construction by using a nickel electro-forming process.

■ Aircraft sleeve-type bearings use nickel for bonding silver to steel.

■ Production of copper sheet directly from copper sulphate solutions for use as a barrier against rot.

■ Nickel plated over plastic airplane propellers to improve abrasion resistance and avoid moisture-caused warpage.

■ Progress made in electropolishing and periodic reverse-current plating techniques.

NICKEL PLATING to specification requirements has led to demand for heavier plates by many British firms. To obtain increased thickness without increasing plant capacity or labor charges, according to E. A. Ollard, *Metal Industry,* commercial platers are using higher current densities of 80 amp per sq foot.

■ Nickel-sulphate, nickel-chloride, boric acid baths, have replaced former nickel-ammonium sulphate electrotype.

■ Solution cannot be used in lead-lined tanks because of solvent action on lead, hence, wood or rubber linings are used.

■ Disadvantages include: Recessed parts difficult to plate at high current densities; rapid changes in bath composition necessitate more rigid control.

GERMAN INDUSTRIAL TECHNIQUES are being carefully studied by American industry. Fortunately German wartime industrial reports are available through the Office of Technical Services. Leaded aluminum alloy bearings were tried. Lead, insoluble in aluminum, provides anti-friction characteristics. This technique has been used by domestic bearing manufacturers for some years, but it is interesting to observe German findings.

■ Best results obtained with alloy containing Pb, 3; Sb, 3 and Mn, 3 percent. This alloy, however, could only be used in cast form. Upon adding 3 percent SN, it became sufficiently ductile to be rolled into a steel backed bimetallic strip. Duralumin backing also used.

■ Aluminum alloys containing 3 to 4 percent cadmium can also be used for bearing materials. Segregation normally encountered in casting Cd-Al and Pb-Al alloys was overcome by special casting methods. Bearing characteristics were improved with high purity aluminum in both types.

SCARCITY OF TIN AND LEAD, plus increased wartime demand for babbitt-surfaced sleeve bearings, caused, according to J. T. Clenny, *Iron Age,* development of new alloys and new babbitting techniques.

■ Babbitts with as much as 83-89 percent tin were replaced by alloys containing 5 percent or less. One of the most successful had percentage composition of: 5 tin, 15 antimony, and the balance lead modified with 1 percent arsenic.

■ To permit increased output of babbitted bearings in sizes over 4 in. O.D., it was necessary to adapt centrifugal casting methods for sizes up to 32 in. O.D.

■ Dove-tailed anchor grooves were source of weakness when bearings were centrifugally cast. Different cooling rates at thin and heavy sections set up thermal

stresses during solidification, promoted segregation and induced growth of hardening constituents.

■ With centrifugal casting, need for mechanical anchorage—both lateral and circumferential—no longer existed. By cleaning steel backing in a molten electrolytic caustic bath prior to babbitting a good bond was obtained.

■ Bearings used on motor turbine and gear sets but not on marine installations where excessive corrosion of the alloy was caused by lubricating oil acid neutralization number greater than one.

■ Centrifugally-cast babbitt quality is a function of rotational speed during casting. If too low, porosity occurred; if too high, segregation. When bearings were water cooled, bearings tinned with pure tin gave better results than those tinned with a lead-tin alloy.

IRON AND STEEL

EXTENDED APPLICATIONS of stainless steel have come from greater realization of its properties and through shielded arc welding techniques. Benefits are in food and chemical processing industries, and in consumer applications. Just announced is a wrought stainless steel with improved corrosion resistance—including sulphuric acid action. Developed by the Carpenter Steel Company, it is first type not cast. Mechanical properties are about same as 18-8 stainless. Result: Processing tanks can be made by forging or other relatively inexpensive forming method.

■ Similar to other austenitic alloys, this stainless cannot be hardened by thermal treatment. Maximum corrosion resistance when heated to 2,100 F and water-quenched.

■ This stainless subject to carbide-precipitation when heated in, or slowly cooled through, sensitizing range. Welded structures need water-quench from 2,100 F. Shot-welded parts need no final heat-treatment.

■ Available as hot-rolled or cold-drawn bars and wires; rounds, rectangles, squares and hexagons; hot- and cold-rolled strip to 8 in. wide and 0.010 to 0.140 in. thick. Tubing also made.

STEEL CASTINGS cannot be ignored as useful engineering materials. Applications cited by J. A. Rassenfoss, *American Foundryman,* include aircraft, railway car and gun mount parts. To show quality of castings, he assembled information on compositions, heat-treatments and resultant mechanical properties, for both static and centrifugal castings. There is one notable reason for using

■ Properties of castings are same in all directions. This is not true of assemblies with rolled or forged parts. Poor transverse properties of wrought stock also deleterious.

SIMULATED SERVICE TESTING of full-sized steel castings show serviceability of finished part and has indicated necessity for consultation between designers and foundrymen. Much grief can be avoided if fillet proportions and section thicknesses are discussed prior to construction of the pattern. Foundrymen can aid in assuring sound castings and minimum weight.

Full-sized testing is not limited to steel casting applications. Fatigue failures in machine parts have been reduced by using simulated tests on full-sized parts.

■ Any combination of repeated tension, compression and torsion can be applied to machine parts by a new fatigue testing machine developed at Research Laboratories Div., General Motors Corp. Failure of traditional test specimens to indicate accurately, prompted design of this machine. Controversy concerning specimen testing as opposed to full-size testing will again spring up.

■ Full-size tests have indicated design changes that stopped fatigue failures in important automotive and airplane motor parts.

■ Metallurgists have long argued that faulty designs, not faulty materials, cause most fatigue failures. Full-sized, simulated service tests may settle the controversy.

ALLOY STEEL SELECTION for high toughness qualities requires all the information obtainable from metallurgists, specimen tests and full-sized, simulated service tests. Selection resolves, according to B. S. Lement, *Iron Age,* into how much toughness is required.

■ When tempered to equal yield strengths, steels that quench completely to martensite are tougher than those that quench incompletely and have all-martensite only at the surface.

■ Designer must consider inter-related needs: Depth of hardening, size of section, severity of quench needed to produce hardening and necessary yield strength.

■ Yield strength falls as the tempering temperature increases. If, to attain high strength, a low tempering temperature is specified, most steels will develop temper brittleness in the tempering range 800-1,200 F.

■ The only answer to the dilemma seems to be in tempering at 1,100 F and water-cooling. This works for small sections. Large sections should be made with a molybdenum-containing steel, which will minimize temper brittleness and give reasonable toughness.

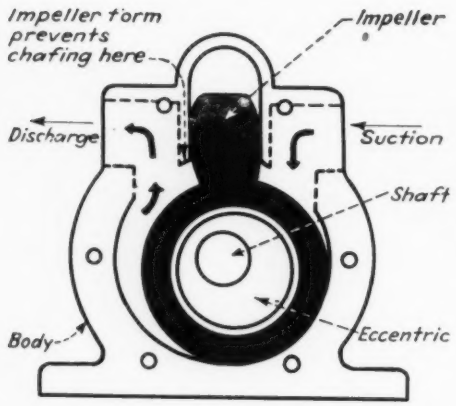
STRESS, an intangible, can be compared to flowing water in a pipe. Weldment failures are so diagnosed by Gerald Von Stroh in *Steel.* Similar to water, stress flows through a structure until it meets an obstacle. In many structures, obstructions are found at weldments.

■ Obstructions can be notches, cracks, hard spots, voids and inclusions in the weld area. Most of these defects can be remedied by properly contouring the weld and careful welding.

■ Some failures attributed to improper welds may be caused by other stress flow disturbances. Machined contours, abrupt changes in direction and section near weld may lead to failure.

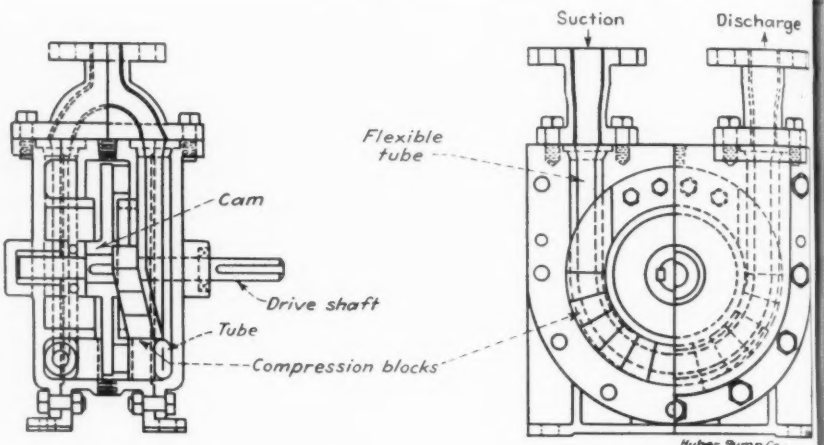
Designs and Operating Principles of Ty

The field of pump design is being bettered and broadened by the use of new



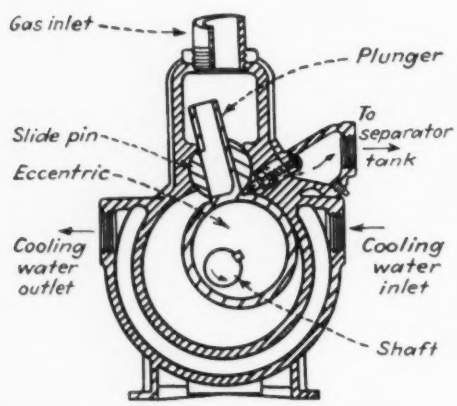
Eco Engineering Co.

9 ORIGINALLY DESIGNED for water use in the marine field, this gearless pump is now available in stainless steel, monel and bronze for handling acids, oils and solvents. The impeller is made of pressure-vulcanized laminated layers of Hycar, 85 to 90 percent hard. Sand, grit, scale and fibrous materials will pass through. With capacities from 1 to 12 gpm and speeds from 200 to 3,500 rpm, pumps will deliver against pressures up to 60 psi. Not self-priming, it can be installed with a reservoir. Operates in either direction and is self-lubricating. Wearing parts replaceable.



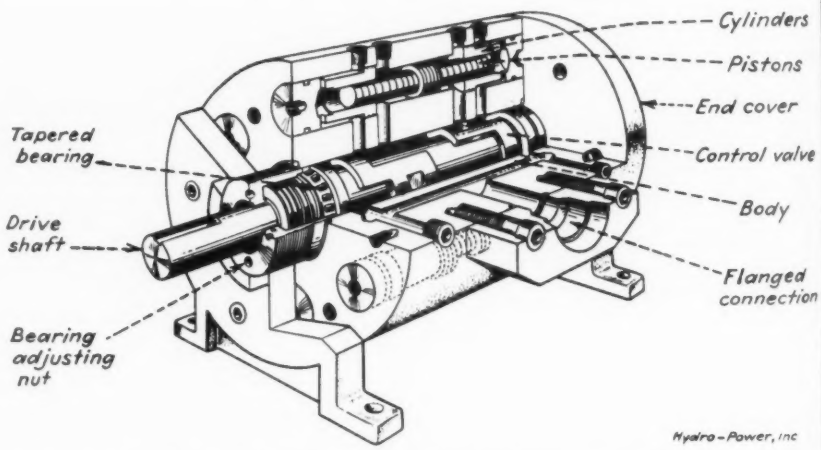
Huber Pump Co.

10 THE SQUEEGEE PUMP consists of a U-shaped flexible tube made of rubber or other flexible material. Acids and corrosive liquids or gases pass through the tube and do not contact working parts or lubricating oil. This prevents contamination of the liquid and avoids corrosion of metal parts. In operation, the tube is compressed progressively from the intake side to the discharge side by cams mounted on a driven shaft. Compression blocks move against the tube, closing the tube gradually and firmly from block to block, which forces the liquid out. As the cam passes the compression blocks, the tube returns to its original diameter. This creates a high vacuum on the intake side and causes the tube to be filled rapidly. The pump can be driven clockwise or counter-clockwise. The tube is completely incased and cannot expand beyond its original diameter. The standard pump is made of bronze and will handle volumes to 15 gpm. The Squeegee develops a vacuum of 25 in. of mercury and will work against pressures of 50 lb per sq inch.



Kinney Mfg. Co.

13 HIGH VACUUM PUMPS operate with rotating plunger action of liquid pumps. Sealing oil lubricates the three moving parts. Parts are accessible without disturbing piping. They are used to rough out system before connecting diffusion pump; to evacuate light bulbs and electronic tubes, and to vacuum dry and distill. Single pumps draw vacuum from 2 to 5 microns; in series to 0.5 micron, and compound pumps draw to 0.1 micron. Pumps can be run in reverse for transferring liquids. Diagonal cored slots, closed by slide pin, form passageway and inlet valve. Poppet or feather outlet valves are used.

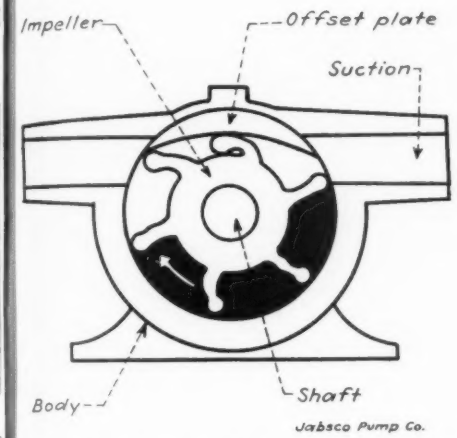


Hydro-Power, Inc

14 A COMPACT MULTI-PLUNGER INTENSIFIER, this hydraulic booster is designed to convert low pressure to high pressure in any oil-hydraulic circuit. No additional pumps are required. Because of the six plungers, the pressure flow from the booster is both smooth and uninterrupted. Many savings in cost are claimed for this booster: First costs are low since high pressure pumps are not required; no operating valves are needed to control the high pressure system, and small cylinder and ram assemblies can be used on operating equipment because the pressure is high. Operating costs can also be low because of efficient use of connected horsepower. The inertia effects of the small operating rams are low so high speed operations can be attained. Hydro-power boosters are built in two standard sizes, each of which is available in two pressure ranges: 2 to 1 and 3 to 1. Volumetric output is in inverse proportion to the pressure ratio. All units have a maximum 7,500 psi discharge pressure. Pistons are double-acting and the central valve admits oil to pistons in sequence and is always hydraulically balanced. Drive shaft integral with valve.

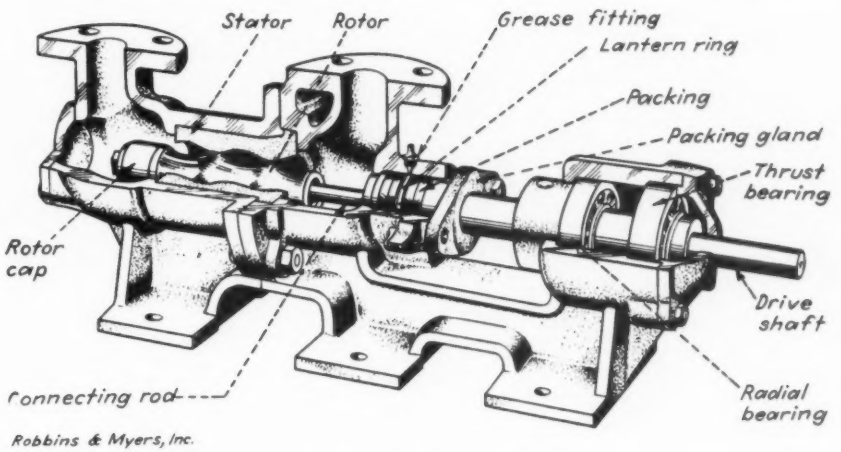
Typical Pumps Used in Industry—II

of new ideas and the application of new materials.



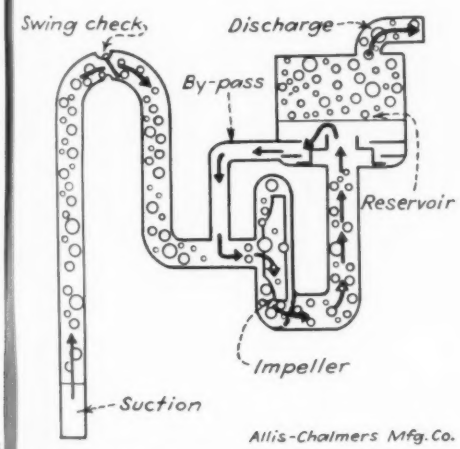
Jabsco Pump Co.

11 DEFLECTED BLADES of flexible neoprene impeller straighten as they leave contact with offset plate. High suction created draws fluid into pump, filling space between blades. Handles animal, vegetable and mineral oils but not napthas, gasoline, ordinary cleaning solvents or paint thinners. Pump operates in either direction and can be mounted at any angle. Pumps run at 100 to 2,000 rpm, can deliver up to 55 gpm, and will operate against 25 psi. Operates at temperatures between 0 and 160 F. Replaceable impeller floats on splined shaft and is not restrained.



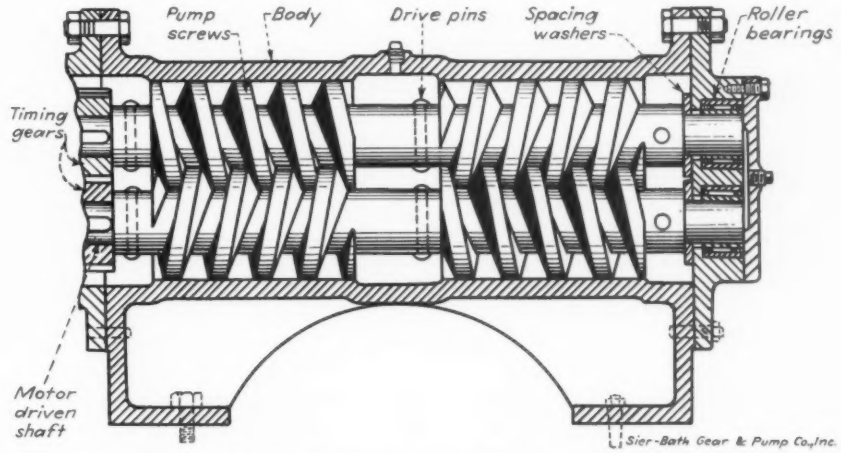
Robbins & Myers, Inc.

12 FOR PUMPING ALMOST ANYTHING from free-flowing liquids to non-pourable pastes, clean or contaminated with abrasives, chemically inert or active, homogeneous or containing solids in suspension, the Moyno is a positive displacement pump that delivers continuous, uniform flow. The one moving part, the low alloy or tool steel rotor, is a single helix and the Hycar or natural rubber stator is a double internal helix. Pumping action is similar to that of a piston moving through a cylinder of infinite length. Containing no valves, this pump will self-prime at 28 ft suction lift. Head developed is independent of speed; capacity is proportional to speed. Slippage is a function of viscosity and pressure and is predictable for any operating condition. The Moyno passes particles up to $\frac{7}{8}$ in. diameter through its largest pump. Pumping action can be in either direction. Largest standard pump, with two continuous seal lines, handles 150 gpm up to 200 psi; multi-stage pumps can pump against higher pressures because there are more seal lines.



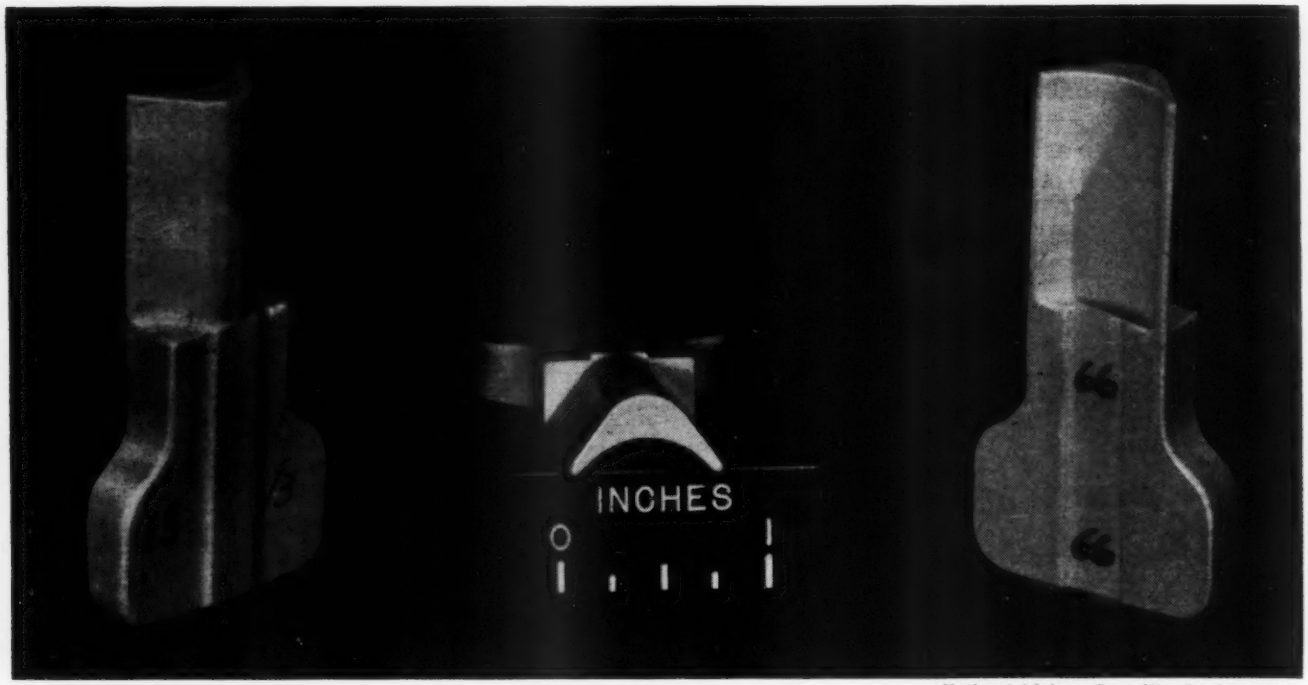
Allis-Chalmers Mfg. Co.

15 SELF-PRIMING PUMP that gives rapid and smooth transition from priming cycle to centrifugal pumping. Pump starts with priming chamber full. Liquid is recirculated through impeller until pump is primed. As priming liquid circulates, air is drawn through impeller and expelled through discharge. When all air is evacuated, discharge velocity closes priming valve completely. Pumps can be supplied with open or closed impellers. Solids up to 1 in. can be passed through 3 in. size pump with open impeller. Self-priming pumps are equipped with a mechanical shaft seal, factory installed and tested.



Sier-Bath Gear & Pump Co., Inc.

16 INTERNAL SCREW PUMPS are very successful for transferring high viscosity petroleum products. They can be used as boiler fuel pumps since they deliver a pulseless flow of oil. For marine or stationary systems, the characteristic low vibration of screw pumps has allowed them to be mounted on light foundations. The absence of vibration and pulsing flow reduces strain on pipes, hose and fittings. The pumping screws are mounted on shafts and take in liquid at both ends of the pump body and move the liquid to the center for discharge. This balanced pumping action makes it unnecessary to use thrust bearings except in installations where the pump is mounted at a high angle. The pumps can be used at any angle up to vertical. Where thrust bearings are needed, anti-friction bearings capable of supporting the load of shaft and screws are used. The intermeshing pumping screws are timed through a pair of precision-cut herringbone gears, which are self-centering, and do not allow side wear of screws while pumping. Pump is designed to give best efficiency under 1,200 rpm for electric motor and 1,300 rpm for steam turbine drives.



National Advisory Committee for Aeronautics

Fig. 1—One of the first ceramic turbine buckets developed for an experimental gas turbine and tested by the National Advisory Committee for Aeronautics. Certain performance tests of these gas turbine blades are reported in N.A.C.A. Technical Note No. 1399

High Temperature Ceramics

Short report on the present state of development of high temperature ceramics. Types of ceramics, physical characteristics and molding methods.

E. L. OLCOTT
Bureau of Aeronautics, Navy Department

RECENTLY DEVELOPED high temperature ceramics show promise for use as molded parts, refractory linings and coatings for gas turbines, jet power plants, rockets and other high temperature equipment. These ceramics seldom replace high-temperature ferrous and non-ferrous alloys but are used to improve their performance.

Interest in high-temperature ceramics is great because further increases in the thermal efficiencies of certain power plants depend to some extent upon improvements in the materials of construction. The thermal efficiencies of these new power plants can be increased if operating temperatures and compression ratios are raised. High-temperature metals show loss in strength and other physical deficiencies at elevated temperatures. On the other hand, certain ceramics, at high temperatures, behave somewhat like metals at low temperatures. For this reason, the use of ceramics in combination with metals, or their future use to replace metals is receiving the attention of many engineers.

Opinions and assertions expressed in this article are the author's and are not to be construed as official or reflecting the views of the Navy Department.

REQUIREMENTS

Commercial porcelain products are made from silicates such as kaolin and steatite. These ceramics are weak and are unsuitable for high-temperature, high-strength use. The requirements for high temperature ceramics are:

1. High melting point
2. Strength at high temperatures (high fatigue and impact resistance)
3. Low specific gravity
4. Resistance to corrosion and erosion by hot gases and corrosive liquids
5. Resistance to thermal shock (rapid variations in temperature)
6. Creep resistance at high temperatures
7. Ease of fabrication and means of attachment

In general, high temperature ceramics are superior to ferrous metal alloys because of their higher corrosion resistance, insulating characteristics, and erosion resistance. They are inferior because they have lower strength, are difficult to fabricate and are fragile. The presence of a stray nut or part in a gas turbine containing ceramic parts is a matter of great concern, whereas a broken metal turbine blade or other part usually does not cause progressive failure in an all-metal power plant.

To realize the advantages of ceramics and high-alloy metals, considerable work has been devoted to their use in combination; ceramics to protect and insulate and metals to provide strength.

FORMS OF CERAMICS

High temperature ceramics are used in paints, coatings and solid form. In paints, they are used to coat large metal surfaces exposed to temperatures not over 1800 deg F. Ceramic paints reduce radiant energy transmission and limit oxidation of the

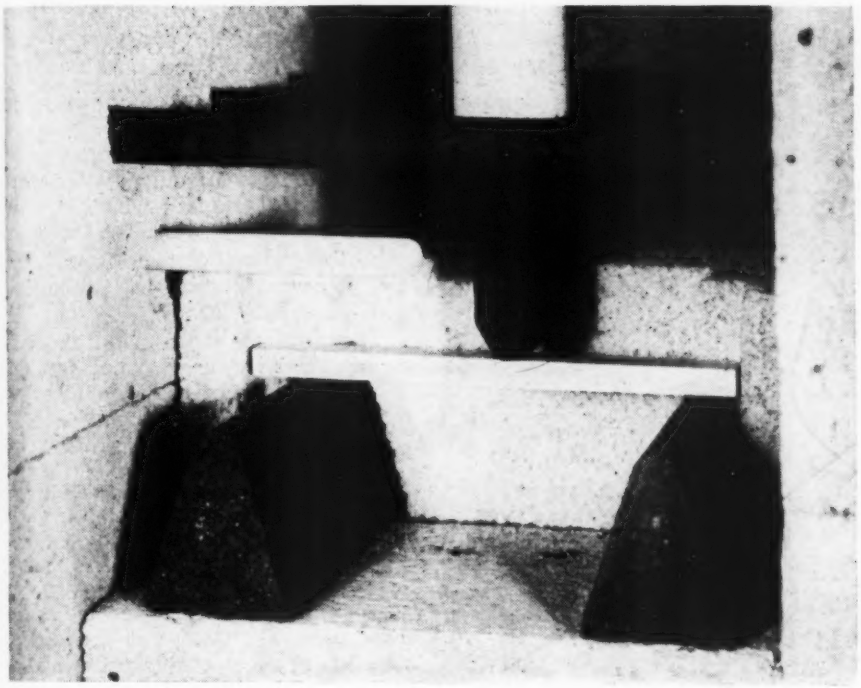


Fig. 2—Typical test set-up for checking the modulus of rupture of ceramic specimens at high temperatures. The white rounded-end rod is a shielded thermocouple.

metal surface. Ceramic paints require no firing and are formulated to be brushed or sprayed and air dried.

Ceramic coatings, on the other hand, are fired on. They are used to protect the base metal, whether low carbon or high alloy steel at temperatures within the plastic range of the metal. There are three types of coatings:

1. Graphites or carbons. Mixtures of graphite and metal powders, fire-polished to oxidize the metal near the surface and to fuse the metal oxide to a continuous film
2. Oxides. Metal oxides such as calcinated magnesia processed in the total absence of water
3. Endothermic materials. Materials which decompose endothermically at high temperatures, such as mixtures of plaster and graphite.

Solid ceramics are cast, sintered or transfer molded. Up to the present, ceramics that retain good physical properties up to a temperature range between 2000 to 5500 deg F are considered satisfactory.

A large number of ceramics have been or are under development. Some ceramic and refractory types showing promise for high-temperature applications are:

1. Metallic oxides (fused beryllia, zirconia and alumina)
2. Metallic oxides bonded with metals
3. Carbides, nitrides and borides bonded with metal and metal alloys
4. Glazed and bonded graphite and carbons.

The physical properties of most importance in high temperature applications are short-time tensile strength and impact at room temperature, and short-time tensile strength, stress-rupture strength, creep rate, endurance limit, and thermal and shock resistance at high temperatures.

Few comprehensive physical data exist for high temperature ceramics. The basic characteristic of all ceramics, particularly the metallic oxides, is their brittleness. In the low temperature range, ceramics break or crack with little indication of plastic flow when subjected to physical shock loads or thermal shock loads. Ductility is low. The stress-strain curve for the majority of high temperature ceramics is straight and terminates at the value for ultimate strength. Ceramics with zero or near-zero porosity have no tensile yield point. Stress-strain curves for porous ceramics resemble that for cast iron. In the high-temperature range, however, these ceramics become somewhat ductile and take on the ductility characteristics of metals at lower temperatures.

The brittleness of ceramics complicates testing procedure. Reliable tensile strengths are difficult to determine by pull tests; instead, the modulus of rupture in bending is used as an indication of strength. (In Fig. 2, a typical setup for the determination of the modulus of rupture at high temperatures is shown.) However, recent investigations on certain ceramics indicate that the tensile strength of ceramics cannot be predicted from modulus of rupture values alone, particularly at high temperatures. Further determination of tensile test data depends on the development of suitable test equipment. In Table I, a number of ceramics now under investigation are listed together with their modulus of rupture at room temperature and at high temperatures. The data given was taken from a number of sources. The significant fact indicated by the values of modulus of rupture shown in the table is that the new ceramics can be considered as structural materials.

Most ceramics creep under the effect of sustained loading at high temperatures. A number of ceramics were tested by the National Bureau

Table I—Strength of High Temperature Ceramics and Refractory Material

Material	Modulus of Rupture—psi				
	Room temp	1800 F	2000 F	2250 F	2400 F
Aluminum oxide bonded with chromium	33,600	.....	.....	.....	12,150
Silicon carbide bonded with metallic silicon	40,000	.....	.....	.....	12,000
Silicon carbide bonded with ferro-manganese-silicon	.....	.....	.....	.....	6,000
Silicon carbide-graphite with glaze of ferro-manganese-silicon	2,000–4,000	2,000–5,000	.....	.....	400–800
Silicon carbide and alumina	2,000–7,000	.....	.....	.....	2,000–3,700
Zirconium oxide	.....	25,000 38,000	25,000 38,000	.....	.....
Beryllia oxide	24,000	.....	.....	4,000	.....

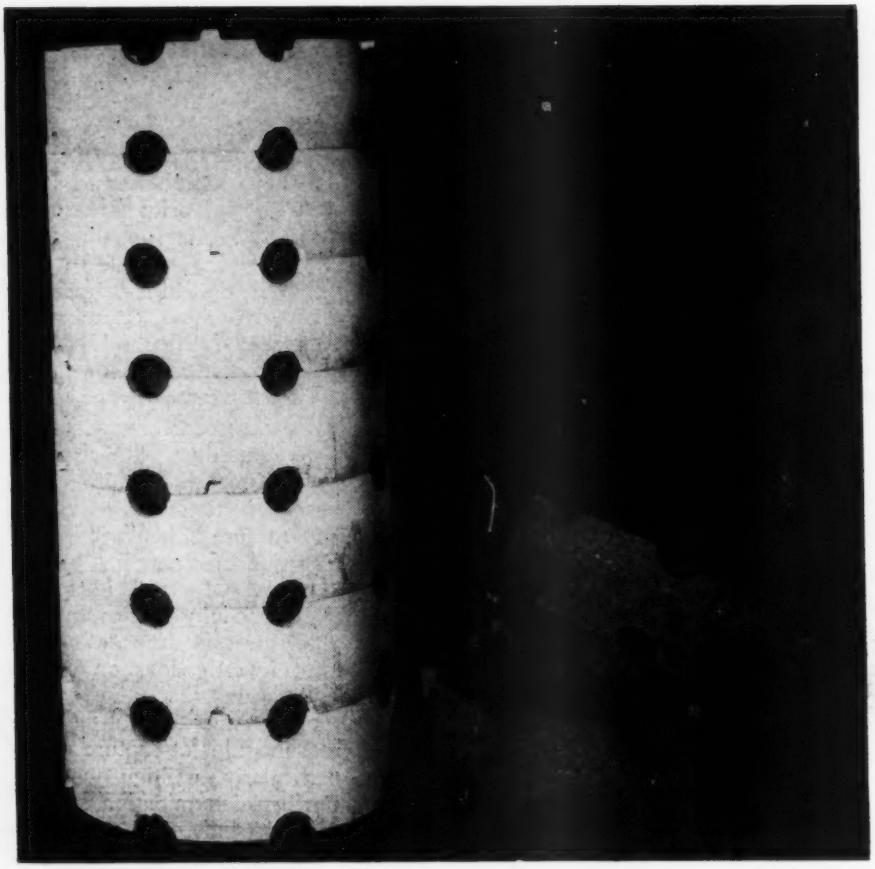


Fig. 3—Molded circular segments tested in a combustion chamber of a jet engine

of Standards. These tests showed that most of them begin to creep at approximately 1600 deg F. The tests showed also that the creep rate is roughly proportional to stress at a given temperature. Fig. 4 shows the approximate range of creep rates for a temperature range from 1500 to 2000 deg. F.

Reliable data showing the performance of high temperature ceramics subjected to thermal shock is not generally available. Ceramics with coarse-grain, loosely-knit structure show up better than dense fine-grain ceramics, but are not as strong. Most of the materials shown in Table I lose strength when exposed to temperature cycling. Tests on several ceramics show losses in strength from 10 to 38 percent. The exceptions are the metallic oxides to which silicon carbide has been added. Silicon carbide increases the thermal conductivity of these and increases their resistance to thermal shock to a remarkable degree.

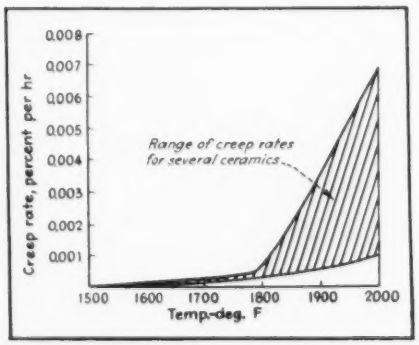


Fig. 4—Range of creep rates at stress-at-rupture for several ceramics at elevated temperatures.

Manufacturing Methods

Production of high temperature ceramic parts has been small. Several production methods are used. Each of these methods includes two steps; the preparation of the ceramic material and the firing process. Before firing the ceramic body is in a "green" state. During firing the body shrinks as much as 20 percent. Consequently any attempt to maintain precise dimensions is difficult to realize in practice.

Several current methods for manufacturing ceramic bodies are:

1. Compacting and sintering—using the techniques of powder metallurgy

2. Slip casting—a mixture or slip is poured into a mold, and dried and fired in the mold

3. Transfer molding—the ceramic mixture is mixed with a thermosetting resin and placed in a separate cavity of a transfer mold. It is made plastic and then introduced in the main mold cavity where it is molded (i.e. —the resin binder is set up). The piece is then taken out of the mold and fired.

4. Hydrostatic molding—spray-dried pellets of ceramic material are put in a rubber mold which is subjected to hydrostatic pressure to compact the material. The compacted part is removed from the mold and fired.

The best molding method depends on the ceramic to be molded, the quantity, size and cost. Usually, further machining is required if the parts have to be held to close dimensions.

Design of Ceramic Parts

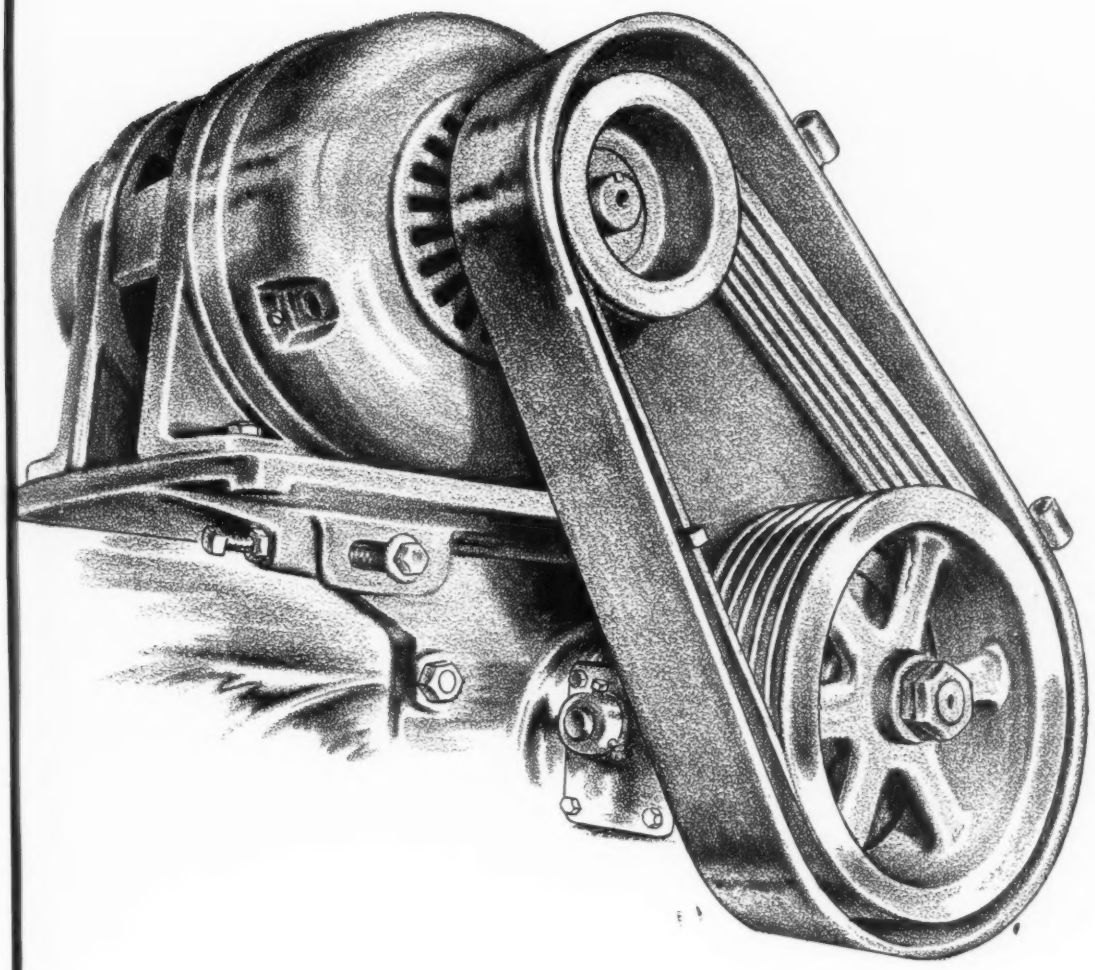
Most high temperature ceramics have little if any ductility at room temperature. Stress raisers such as sharp corners, notches and holes in highly stressed areas should be avoided. Attachments for ceramic parts can be designed to allow for some degree of freedom to avoid restraints that will introduce undesirable loads if the ceramic part creeps or expands. Tolerances on dimensions should be generous in order to reduce machining operations.

On coated parts, ceramic coatings thin out at sharp corners; therefore it is necessary to know something about the thickness of the coating applied, and the way in which the thickness varies, in order to dimension the uncoated part properly.

In common with metal die castings, ceramic parts, with a high ratio of surface area to volume, show higher strengths because of skin effect. The continuous surface film has greater structural continuity than the material within the piece. Because of this, physical properties obtained from tests of small specimens must be examined carefully before applying the data to the analysis of larger pieces.

The strength of ceramic parts can be proved only by service tests under actual operating conditions; but recognition of the physical deficiencies enables the designer to exercise elementary precautions during preliminary design.

Although the latest developments in high-temperature ceramics leave a lot to be desired, continuing development of existing types will bring about a gradual improvement in physical properties. To find wide use for elevated temperature applications, the physical properties of ceramics at high temperatures should approach the low temperature physical characteristics of the metals they replace. And must show more resistance to thermal shock.

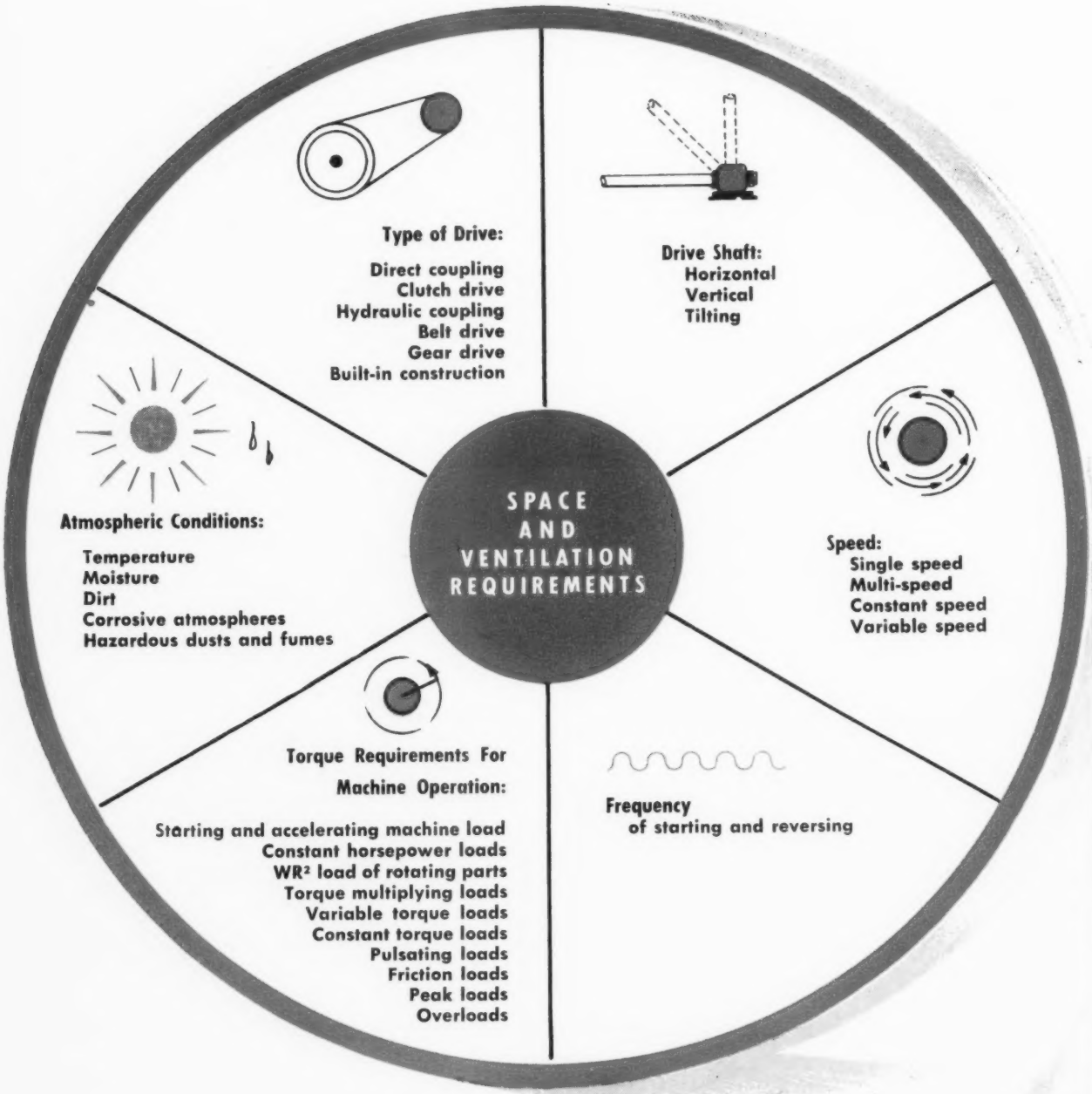


MOTOR DRIVEN MACHINES

MECHANICAL ARRANGEMENTS FOR BEST MOTOR PERFORMANCE

Design of machine drives to obtain efficient motor operation and to make possible the selection and use of the most economical type and rating of motor.

Drive and Load Factors That Affect Motor Selection



Requirements for Space and Ventilation Inclosures for Ambient Conditions

Space needed for mounting, installing, maintaining and ventilating a motor is frequently overlooked. To make certain that the space provided for these purposes is ample, designers should check space with frame size dimensions as given in standard dimensions sheets. These sheets can be obtained from any motor manufacturer.

Space requirements for motors of given horsepower rating vary with the speed. High speed motors are smaller than low speed motors. Motor frames are designed to produce a given torque. The frame size increases with torque. To conserve space, therefore, keep motor speeds as high as possible. High speed motors cost less than low speed motors of the same horsepower.

Vertical motors cost more than horizontal motors, therefore, the designer should strive for horizontal installation wherever possible.

Space limitations may affect the choice of motor. When space is limited, designers should consider built-in motor construction, gear-motors, forced ventilation using external blowers, or a smaller frame motor with Class B insulation that permits higher temperature rise.

Atmospheric conditions are governing factors in the selection of type of motor inclosure. In selecting type of inclosure, the designer should check first for ventilating capacity. He should then determine whether or not a cost differential would justify a change in the motor location or a change in the type of motor.

For ordinary industrial conditions, use open motor.

When not sufficiently protected by external covers from splashing liquids, use splashproof motors.

When motor operates in air filled with moisture or gaseous fumes, use totally inclosed, fan-cooled motors, or open motors with special insulation. Specially insulated low-voltage designs may be necessary.

When exposed to flammable gas atmospheres, no more hazardous than gasoline, use Class I, Group D motors.

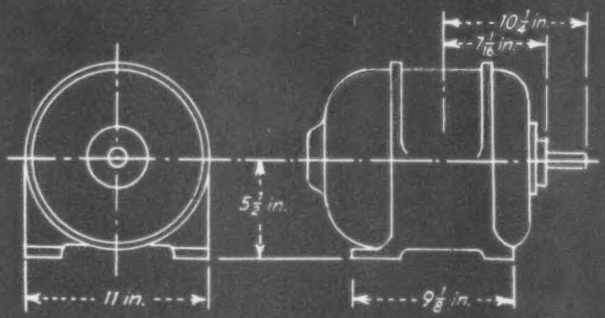
When exposed to combustible dust atmospheres, no more hazardous than flour, use Class II, Group G motors.

When installation is outdoors in all climates, use totally inclosed, fan cooled motors. In mild climates, splashproof motors may be adequate.

Ducts of ample area to supply cool air should be provided in machine compartments and unit housings that inclose motors. Limited ventilation may cause motor burnouts.

Temperature-rise ratings of motors assume that the normal ambient temperature is not more than 40 C. For ambient temperatures exceeding 40 C, Class A insulated motors with lower temperature rise may be used provided the total temperature (ambient plus rise) does not exceed 90 C for open and 95 C for inclosed motors. These limits may be raised to 110 C for open motors and raised to 115 C for inclosed motors by using Class B insulation.

TORQUE DETERMINES PHYSICAL SIZE OF MOTOR



FRAME 225, TOTALLY INCLOSED, FAN-COOLED

1200 rpm
2 horsepower

1800 rpm
3 horsepower

3000 rpm
5 horsepower

TORQUE PRODUCED BY FRAME 225

TORQUE OF MOTOR IS FIXED BY ITS MASS OR SIZE, THIS IS BASICALLY CORRECT FOR SPEED GREATER THAN 600 RPM. TORQUE IS A FUNCTION OF VENTILATION IN SO FAR AS VARIATIONS ARE CONCERNED.

Couplings and Clutches

Driven machine should always be coupled directly to motor shaft when machine speed is the same as the motor speed, provided that the motor starting torque is sufficient to start the machine with its connected starting load.

Flexible couplings of many types are satisfactory. In selecting a coupling, factors that should be considered are: Size and space requirements, degree of shaft misalignment permitted, initial cost and maintenance.

Splined couplings readily permit withdrawing the motor for servicing in applications where a coupling between motor shaft and driven shaft is inaccessible.

Shear pin couplings protect both the machine and the motor from excessive overloading when the possibility exists of a machine jamming.

Built-in construction offers many advantages where direct connection is possible. Stator shells and shaftless rotors of standard motors frequently can be incorporated in a machine with savings in space and cost.

Clutches may permit use of smaller motors when torque required to start a machine is much larger than that required when machine is running, although a high torque motor of no larger frame size may be available.

Clutches used to bypass starting loads sometimes permit motor starting across the line, thus, simplifying the starting equipment.

Mercury clutches and clutches of the centrifugal type installed between the motor and its load provide automatic torque control and overload protection. In operation, the inertia delay of the actuating mercury or parts retards the engagement of the friction members long enough for the motor to reach full speed before it picks up the load. The motor is thus able to exert its running torque to start and accelerate the machine load.

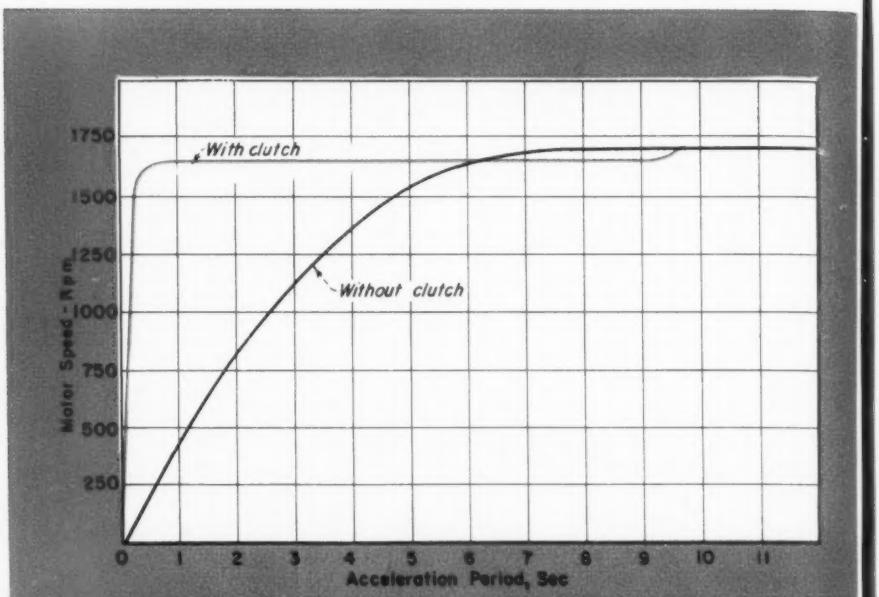
Compressed air clutches of the expanding gland type not only permit the motor to come up to speed before the machine load is applied, but also can be controlled to exert different amounts of grip between the clutch members and thus control the torque delivered by the motor. Shock loads are cushioned by air-filled gland, thus relieving the machine and motor from excessive strains.

Multiple metal disk clutches and those of the dished plate type permit some latitude of slip in starting and under overload. The speed of pick up depends upon manual operation.

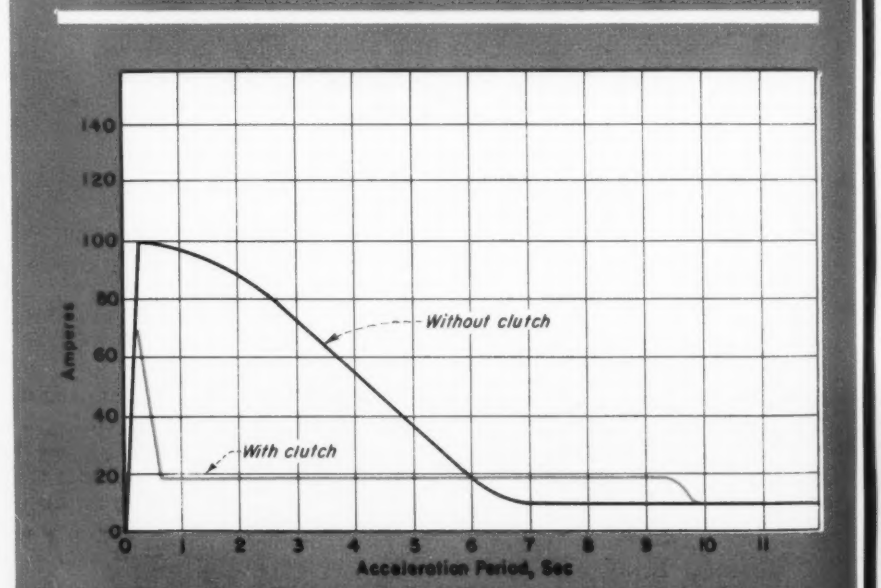
Rapid deceleration and quick stopping can be obtained with mechanical brakes, magnetic brakes, dynamic braking, or plugging.

Motors with built-in brakes are intended to be used for holding loads and for stopping the motor rotor, but not for lowering or stopping loads. On cranes and hoists, therefore, the brakes in such motors are to be used only as an auxiliary to automatic mechanical load brakes, other mechanical brakes, or to dynamic braking or plugging control.

Quick stopping may be essential for other than mechanical or operational considerations. The designer should decide whether quick stopping is necessary to prevent serious injury to the operator in the event that his arm or clothing should be caught in the machine.



Motor speed with and without a delayed action clutch.



Power required to start and accelerate a high inertia load with and without a delayed clutch.

Belt and Chain Drives

Belting is a simple, inexpensive and efficient form of drive for machines in which it is not necessary to maintain an absolutely fixed ratio between the motor and the driven shaft.

Under heavy overloads, belts will slip on the pulleys. Because of their inherent ability to stretch under load, they will absorb a portion of the shock of suddenly applied loads, and thus protect to some extent both motor and driven machine.

V-belts and flat belts made of leather, fabric, and rubber compositions, and specially tanned or waterproofed leather belts, can be selected to meet practically any atmospheric condition. Space required and maintenance costs of belt drives are usually more than for other types of mechanical transmissions.

Belt tension adjustment is a constant problem. Sliding bases are almost always used to obtain belt adjustment.

Pivoted motor mountings, though in some designs are difficult to include, take up the slack as the belt stretches under load and thus reduce slip.

Drives in which the belt is horizontal, or not more than 45 deg from the horizontal, usually perform satisfactorily. When the lower side of the belt is the tight or driving side, the sag of the upper side increases the arc of belt contact on the pulleys. Mounting one pulley vertically above another should be avoided, especially when the lower pulley is the smaller.

CENTER DISTANCE. A short center distance is unsatisfactory. With a short belt, the rate of flexing each cross-section per revolution may be great enough to destroy quickly the resiliency of the belt. Shortening the center distance reduces the arc of contact on the smaller pulley, thus reducing the amount of power that can be effectively transmitted.

Minimum center distance should be at least two and a half times the diameter of the large pulley, with from three to five a better value.

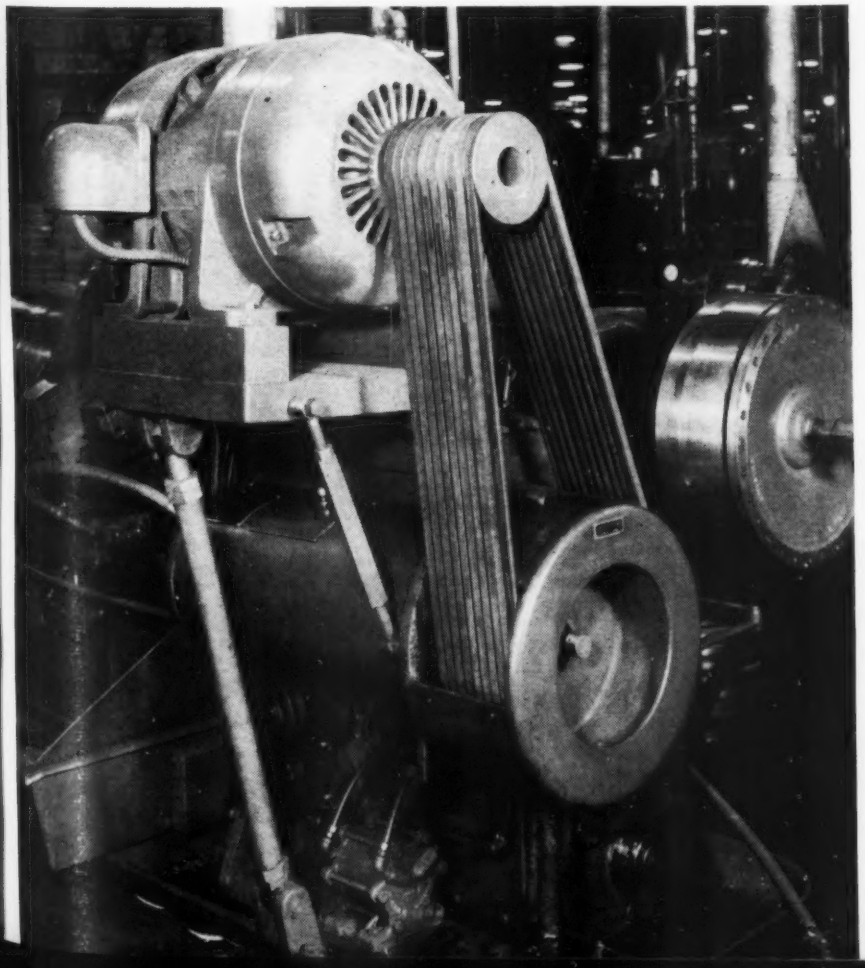
Width and thickness, or cross-section, of belting should be determined on basis of peak loads and possible overloads, which do not necessarily correspond with the horsepower rating of the motor.

BEARING PRESSURES in the motor and deflection of the motor shaft result from belt pull. Belt pull is a function of power transmitted, pulley diameters, and arc of contact. Application of standard motors having two bearings, of course, is limited by considerations of bearing pressure, bearing speed and shaft deflection. Bearing pressure increases as the pulley diameters become smaller.

Three-bearing construction to obtain additional bearing surface may be an advantage where abrasive dust and grit are present.

Maximum bearing loads for a few standard frames with two sleeve bearings are:

Frame	Max load, lb
B-204	80
B-224, 225	125
B-254	200
B-284	250
CD-73–75	475
CD-83–85	600
CD-93–95	750



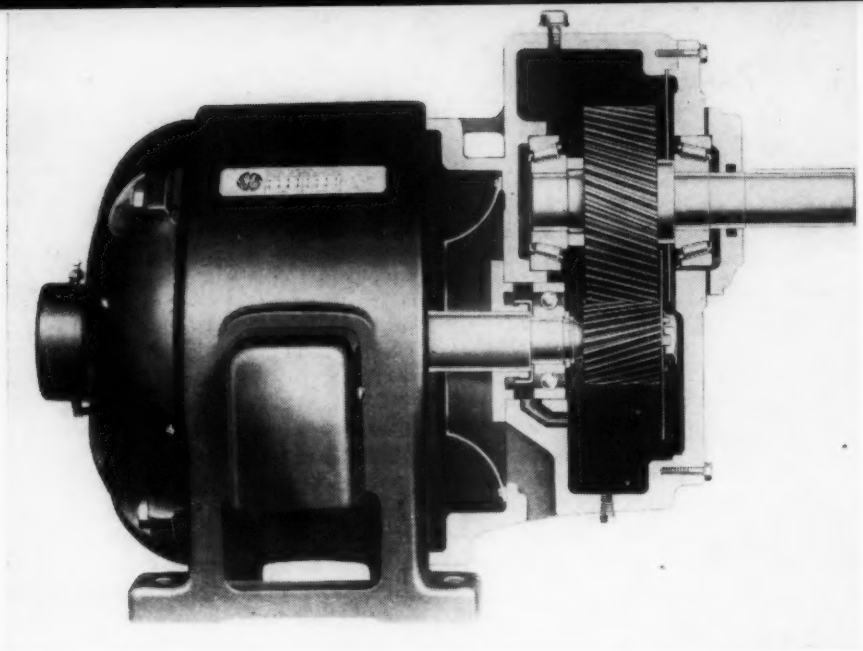
CHAIN DRIVES. Chain speed, silence of running, wear life and freedom from maintenance will vary according to the combination of pitch, sprockets, and chain width.

Pitch of chain should be selected to give smooth and silent running in chain drives for business machines, moving picture projectors, bakery and laundry equipment.

Speed of the faster running shaft establishes the maximum pitch that should normally be used. The number of strands of roller chain or the width of the toothed or "silent" chain is determined by power to be transmitted.

Speed of either silent or roller chains in feet per minute is the number of teeth in the sprocket times the chain pitch in inches times the revolutions per minute divided by 12. For the same number of teeth and revolutions per minute, the chain speed is directly proportional to the pitch.

Service factors can be furnished by engineers of the chain manufacturers by which rated capacities must be multiplied to conform to service conditions such as lubrication, shock or pulsating loads, position of the drive, intermittent or continuous service, distance between sprocket centers, speed ratio and similar variables.



Gear Drives

When determining the safe duty limits of a gear train, the designer should not neglect to recognize that the duty required of gear trains in machines and equipment is different than that required of motors.

Gears may be abused by excessive loads of any duration, whereas the continued operation of the motor is not greatly affected, except by excessive duration of overloads.

Gears must be designed on the basis of working stress developed during shock and stalled loads, or with a specified maximum variation in load, rather than on the normal full-load torque of the motor. Consideration must also be given to "hunting" caused by the flywheel effect of high inertia loads, or the recurrence of the high point in a varying cycle on the same tooth.

Gears can be driven in either direction of rotation. But if so, the designer should specify minimum backlash required and minimum end play in the bearing assembly.

Bevel gears, spiral gears, helical gears, and worms require thrust absorbing arrangements to resist the thrust components of load.

Herringbone gears provide continuous tooth contact without axial thrust. Herringbone gears give smooth operation, are efficient in performance and have long life.

To avoid excessive bearing pressures and shaft deflections in motors, minimum diameters of pinions have been established by motor manufacturers on the basis of power transmitted, the speed and the tooth angle.

Maximum pitch-line speed for steel pinions is about 1,300 ft per min.

A high speed motor geared down to machine speed may be less costly than a directly coupled, low speed motor. More maintenance may be required than for direct coupling, but maintenance is less than that required for a belt drive. Less space is needed for a gear train than for a belt drive.

Fixed speed, geared drives cost more than belt drives, but they are more compact and allow the selection of any speed ratio.

In single reduction and the first stage of double reduction worm gearing, where high speed is likely to create noise, a laminated plastic gear is often used with a hardened steel worm. In the second step of the double reduction where high torque transmission is important and noise caused by speed is not likely to exist, metallic gears are used in conjunction with a hardened steel worm.

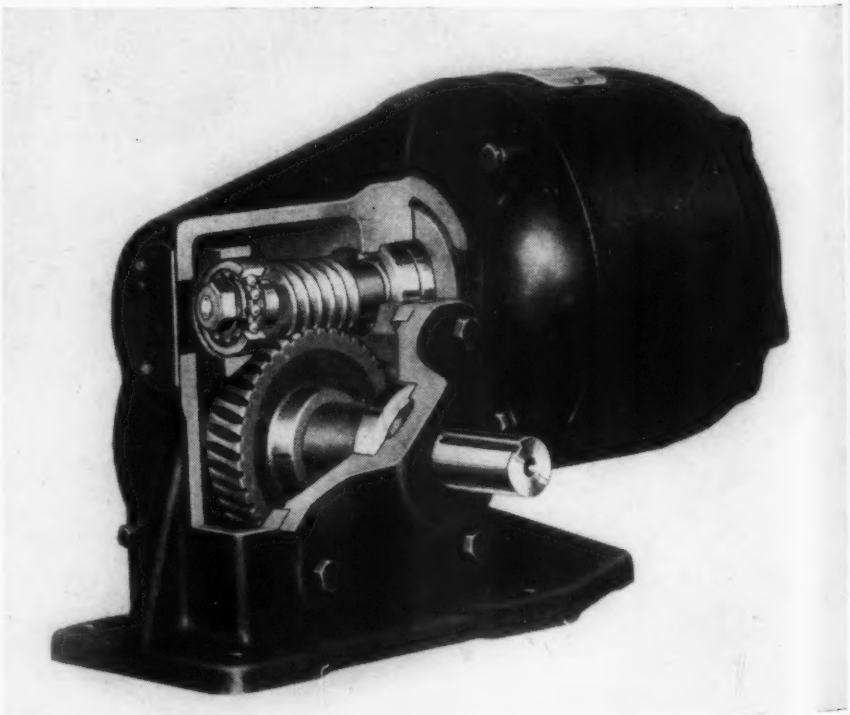
Gear-motors, that is, motors with a built-in speed reducer, provide compact drive arrangements. Gear-motors are built in several types of mechanical constructions with speed ratios of 3 to 1 upwards. Capacities range from fractional horsepowers to more than 50 horsepower.

Gear-motors operate without mechanical slippage and can be installed in less space than is required for a belt and pulley drive.

For output shaft speeds of 600 rpm and less, gear-motors may be more economical and efficient than belt drives.

When specifying the torque desired in a gear motor, the designer should state whether the torque is that developed at the motor shaft or at the gear motor output shaft.

Misalignment with consequent gear wear can be reduced to a minimum by providing a stiff mounting for the gear motor as close as possible to its gear shaft extension.



Hydraulic Couplings

Hydraulic couplings transmit power derived from the mass and velocity of the fluid set in motion on the prime mover side of the coupling. Prime mover and driven machine are not connected mechanically.

Motors and driven machines can be directly connected by hydraulic couplings or through belts, gears, chains, or speed reducers.

Hydraulic couplings are not designed to correct misalignment. They must be installed with reasonable care and accuracy.

Installed between motors and driven machines, hydraulic couplings make the maximum running torque of the motor available for starting and accelerating heavy inertia loads.

At starting, since no mechanical connection exists between the motor and the driven machine, the load on the motor is only the shear resistance of the hydraulic fluid. As the motor speed increases, the hydraulic torque builds up at a slower rate and uniformly accelerates the machine load up to its running speed.

After starting, the motor reaches its peak torque speed in a few seconds. The motor then retains this speed. No conditions of operation can pull the motor below this speed.

Output torque is always equal to full input torque. Slip of 4 to 5 percent is inherent in the coupling at normal operating load and speed, so that the maximum power efficiency is about 95 percent or better, but torque transmission is always 100 percent.

Current consumption falls rapidly from the initial starting current as the motor speed increases to peak torque speed. Motor heating and power demand are thus minimized, and motor control circuits are simplified.

At peak torque motor speed, full motor torque is available for any machine torque from zero to 100 percent.

OVERLOAD PROTECTION. Overloads slow down the driven member of the coupling and the excessive torque of the overload is absorbed by the stored WR^2 energy of the rotor.

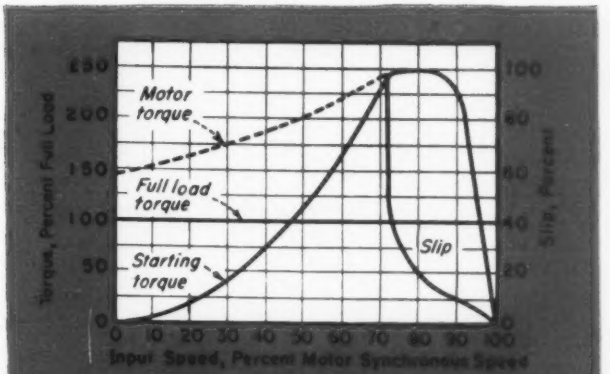
Shock loads from the driven machine are smoothed out and cushioned before they reach the motor.

Shock waves never strike solid resistance, therefore, driven machine and motor are not subjected to severe mechanical strains.

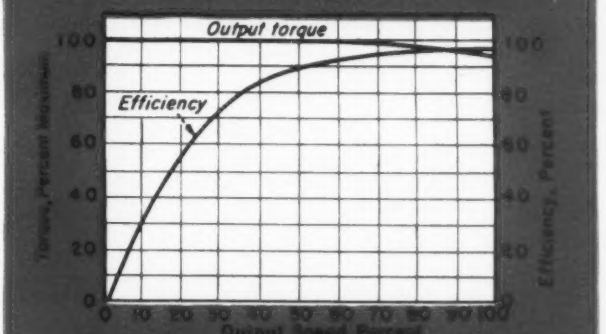
With a stalled output shaft, the motor speed decreases to maximum torque speed and the torque increases correspondingly. The excess current supplied at this speed trips the overload electrical controls before the coupling becomes overheated.

Torsional vibration of the motor or driven machine is dampened out by the coupling.

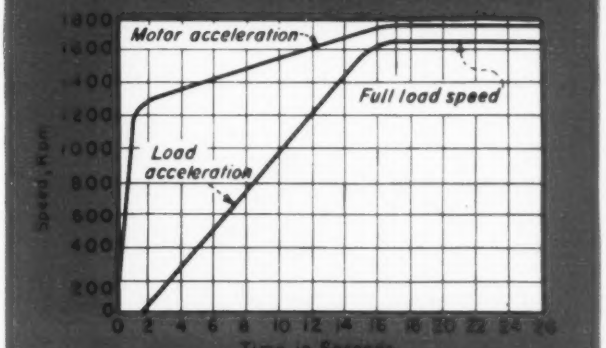
Coupling size is normally selected to give from 3 to 5 percent slip at full running load and full speed.



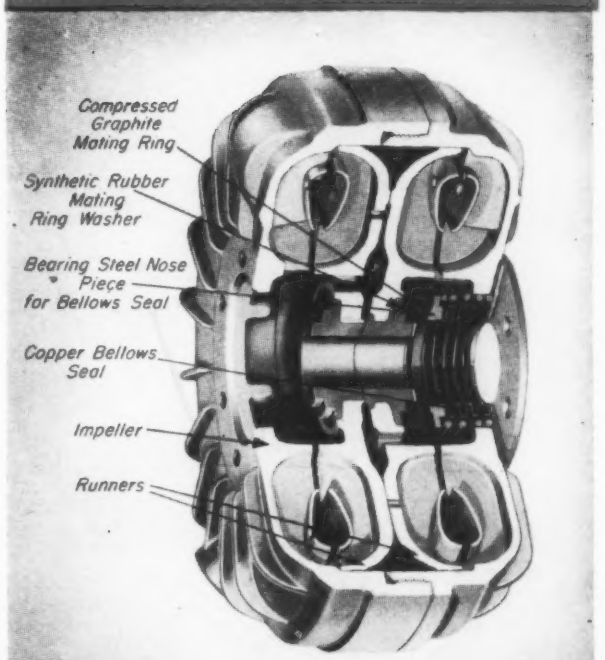
Resistance of hydraulic coupling to starting of motor is small with output shaft at rest.



Output torque of hydraulic coupling is practically equal to its full input torque.

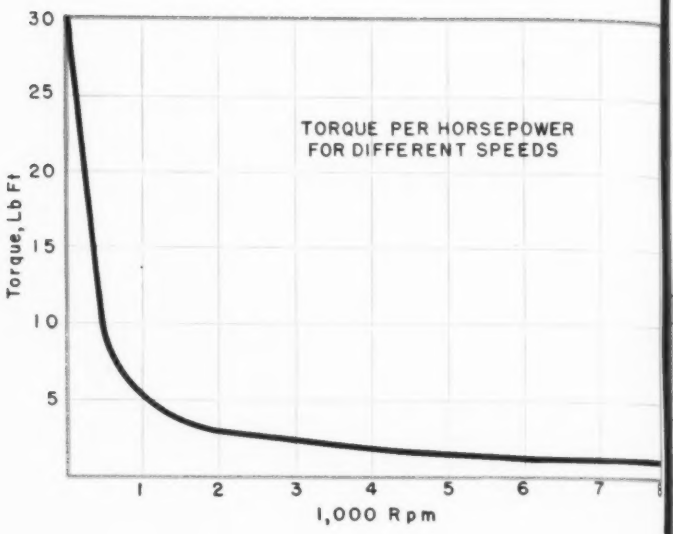


Performance of hydraulic coupling and motor when accelerating a load to full speed.



Hydraulic coupling has no mechanical connection between power source and driven machine. Incoming power rotates the impeller, drives the runner, and is delivered at output shaft.

Torque and Horsepower



Turning moment exerted by a motor is expressed in terms of torque. It is the product of a force and the perpendicular distance from the line of action of the force to the axis of rotation. Torque is usually expressed as pounds of force acting at a radius of one foot.

Where

P = force acting, lb
R = radius at which P acts, ft
T = torque, lb-ft
$T = PR$

Horsepower is work done per unit of time. One horsepower equals 33,000 ft-lb of work per minute. When work is done by a source of torque to produce M rotations about an axis, the work done is

$2\pi RPM$ or $2\pi TM$

When rotation is at the rate of N rpm, the horsepower H delivered is

$$H = \frac{2\pi RPN}{33{,}000} = \frac{TN}{5{,}250} \quad (1)$$

or

$$T = \frac{33{,}000H}{2\pi N} = 5{,}250\,\frac{H}{N} \quad (2)$$

$$= 63{,}025\,\frac{H}{N}\text{, in.-lb} \quad (3)$$

When work is done to move a load of P pounds through a distance S feet in t minutes, the horsepower H expended by the prime mover is

$$H = \frac{PS}{33{,}000t} \quad (4)$$

Example: A drilling machine requires 12 tons feed pressure and the feed speed varies from ½ to 6 in. per min. No load rapid traverse speed is 250 ft per min. Frictional load at all speeds is 600 lb and for practical purposes is essentially constant.

Horsepower required for feed at maximum speed is

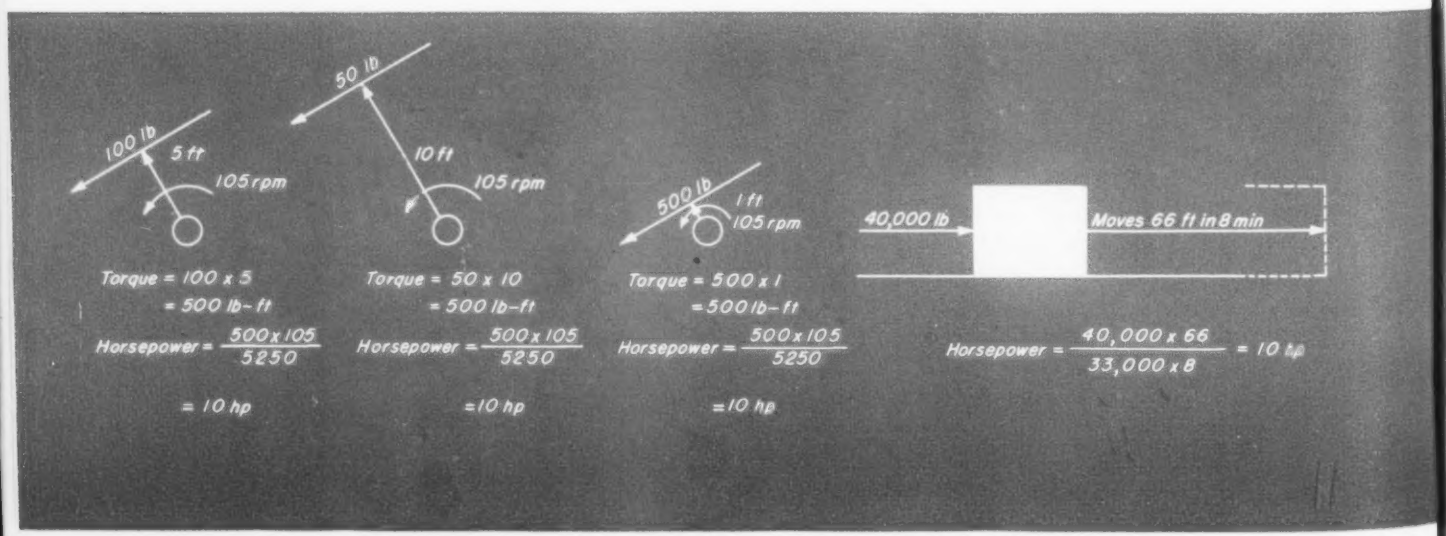
$$H = \frac{(12 \times 2{,}000 + 600) \times (6/12)}{33{,}000 \times 1}$$
$$= 0.373 \text{ hp}$$

Horsepower required for rapid traverse is

$$H = \frac{600 \times 250}{33{,}000 \times 1} = 4.55 \text{ hp}$$

Instead of using one 5 hp variable speed motor to drive both feed and traverse, it would be much more economical and satisfactory for the designer to use a ⅓ hp variable speed motor for the feed, and a one speed 5 hp motor for the traverse. A 5 hp variable speed motor would cost more than a ⅓ hp variable speed motor plus a 5 hp one speed motor.

In addition, a 5 hp variable speed motor would have a smaller speed range than a ⅓ hp variable speed motor.



Multiplication of Torque

In a motor-driven pulley or gear train running at constant speed, disregarding frictional losses, the horsepower delivered by the motor equals that of the output shaft. But the torque imposed on the motor shaft may be greater or less than the torque on the output shaft of the train. Which torque is greater depends upon the ratio of the speeds of the shafts. Where

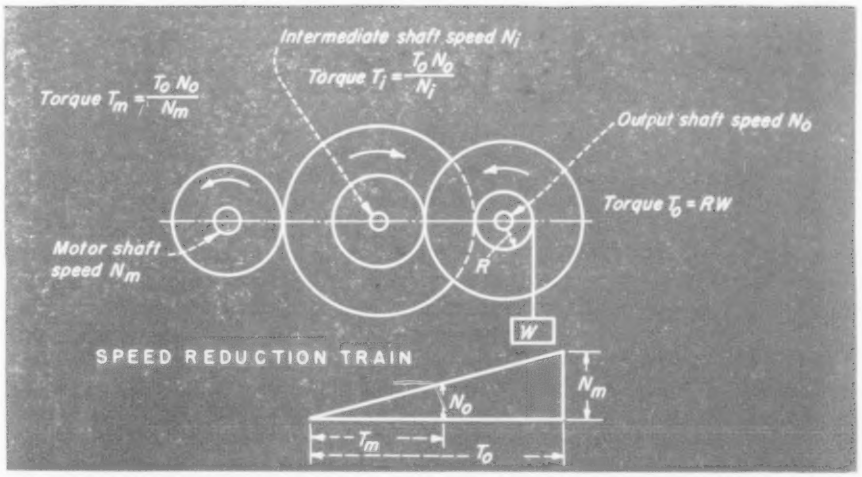
H_m = horsepower delivered by motor shaft
H_o = horsepower delivered by output shaft
T_m = torque on motor shaft, lb ft
T_o = torque on output shaft, lb ft
N_m = speed of motor shaft, rpm
N_o = speed of output shaft, rpm

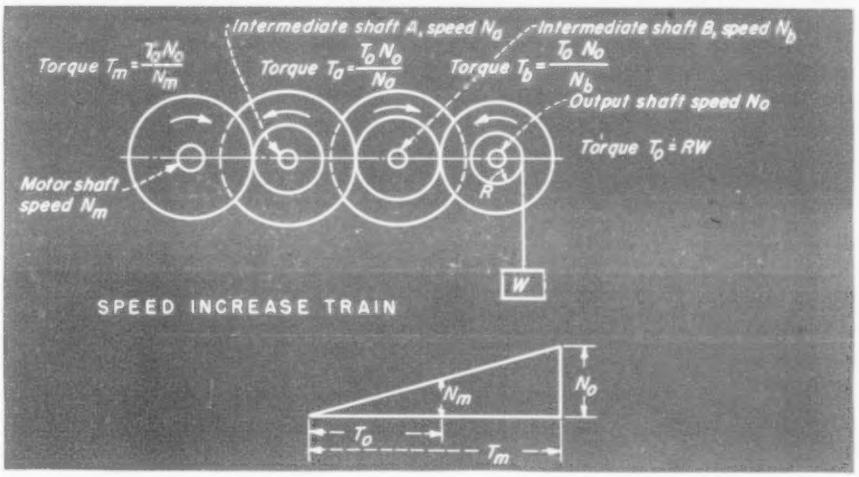
$$H_m = \frac{T_m N_m}{5{,}250} = \frac{T_o N_o}{5{,}250} = H_o$$

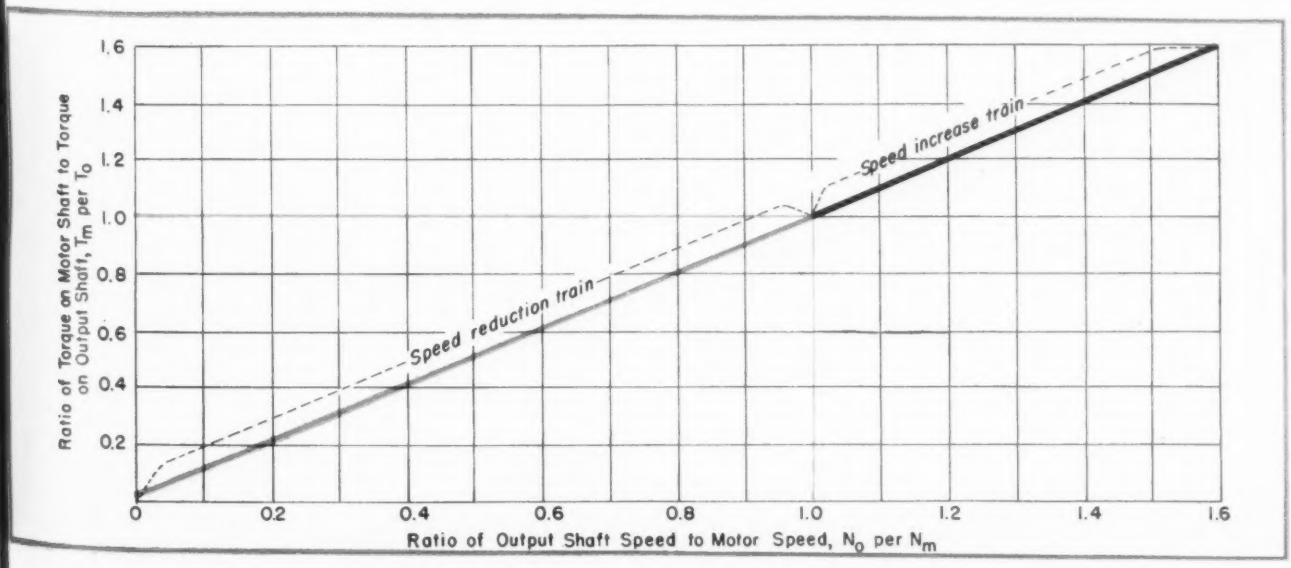
therefore the torque T_m at the motor shaft that results from a torque T_o at the output shaft is

$$T_m = \frac{T_o N_o}{N_m} \qquad (5)$$

These relations between horsepower and the effects of introducing speed changes in a drive, such as the use of speed reducers, are important considerations in selecting a motor that will produce a desired torque.







Friction Loads

Mating surfaces of bearings, sliding parts and gear teeth may offer static friction forces when starting a machine that are many times greater than the friction forces present in the machine when running.

In machines that have been idle for several hours or more, thin lubricant may drain from the mating surfaces, destroy the oil film and promote the formation of dry spots, and thus cause the mating surfaces to be only partially lubricated when the machine is started.

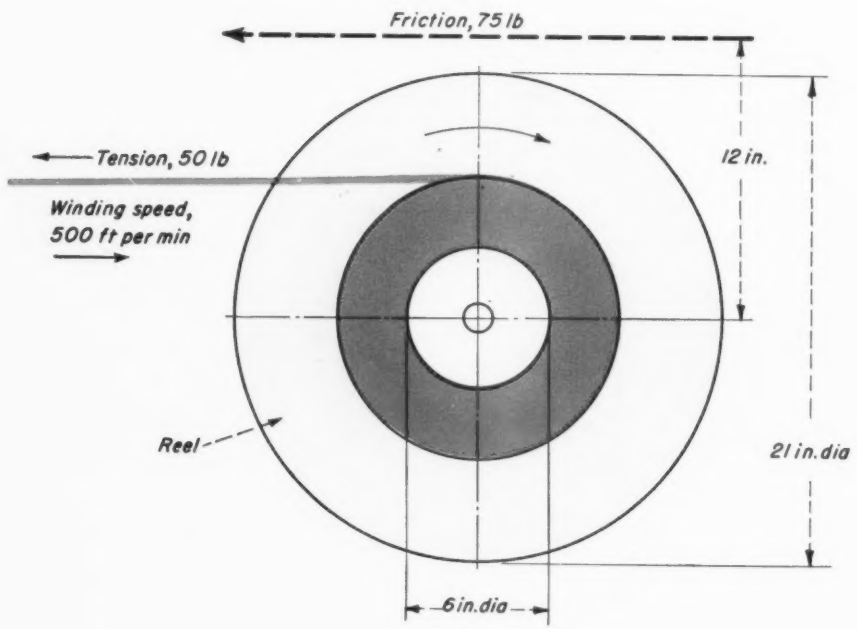
Lubricants lose some of their ability to flow when cooling from running to ambient temperatures. The capacity of a lubricant to form an effective oil film decreases with temperature.

To obtain low breakaway torque of the machine and thus ease motor starting in direct connected machines, it is important that friction forces be reduced as much as possible through the use of low-friction materials and bearings and efficient lubricating methods.

Friction forces in machines operating at constant running temperature are practically constant regardless of speed. But the power required to overcome the torque of friction loads increases directly with speed.

Before selecting a motor to drive a variable speed machine, especially when the work load is small in comparison with the friction load, it is highly desirable to determine the friction torque of the machine by test.

In many machines such as winding reels and spoolers that operate at variable speeds and a constant work load, the horsepower required for the work may be constant, but the horsepower required to overcome friction torque in the machine will increase directly as the speed.



EXAMPLE: A variable speed reel with a 6 in. dia winding roll winds material at the rate of 500 ft per min under a tension of 50 lb. Rolls up to 21 in. dia are wound. Friction torque on the reel is 75 lb-ft. Speed of reel varies from 318 to 91 rpm to maintain constant linear winding speed.

Horsepower required when starting to wind on empty reel:

For tension [From Eq (1)]

$$H = 50 \times \left(\frac{6}{2 \times 12}\right) \times \frac{318}{5{,}250} = 7.55$$

For friction [From Eq. (1)]

$$H = \frac{75 \times 318}{5{,}250} = 4.54$$

Total hp required at high speed

$$H = 7.55 + 4.54 = 12.09$$

Horsepower required when reel is full:

For tension [From Eq. (1)]

$$H = 50 \times \left(\frac{21}{2 \times 12}\right) \times \frac{91}{5{,}250} = 7.55$$

For friction [From Eq. (1)]

$$H = \frac{75 \times 91}{5{,}250} = 1.30$$

Total hp required at low speed is

$$H = 7.55 + 1.30 = 8.85$$

Note that the horsepower required to overcome tension is the same at high as at low speed, but that 3.24 hp more is required to overcome the friction torque at high speed than is required at low speed.

Bearings and Lubrication

Bearings for supporting shafts should be widely spaced to give positive alignment, especially when loads are overhung.

Long running fits at bearings help in preventing entrance of dust, dirt, water and corrosive materials.

Cartridge-type bearing housings exclude dirt even when removed from a disassembled machine.

Prelubricated bearings can be used in machines at places where access for lubrication is difficult.

Friction of rolling contact bearings of the ball, needle, straight roller, and tapered roller types, is not only low, but is almost entirely independent of speed, load and temperature. Starting friction is not much greater than running friction.

Porous bearings of sintered mixtures of copper, tin, and graphite powder, when impregnated with oil will operate for long periods of time without further lubrication.

Bearing seals keep dust and grit out of the bearing and thus protect the bearing and journal surfaces. These seals also retain oil in the bearing.

Sleeve bearings permit considerable axial movement of the shaft when there is no interference from shoulders. Misaligned pulleys will cause shafts to weave, if shafts are shouldered the shoulders may quickly pound and upset the ends of sleeve bearings.

For horizontal operation, motors of sleeve or ball bearing type may be used. In applications where slight end thrust occurs, ball bearings are preferred. Because of lubrication difficulties ball bearings are preferred when motor operates in a vertical position.

THRUST AND END PLAY. Direct connected machines such as propeller-type fans and deep well pumps may impose axial thrust on motor bearings.

Standard ball bearing motors may have sufficient capacity to take moderate end thrust. Heavy machine thrusts may require plate type oil lubricated bearings.

Machines in which end play must be held to extremely small amounts may necessitate the use of preloaded ball bearings or sleeve bearings with end play limiting arrangements.

On machines that operate under extreme variations of temperature, where temperature changes cause appreciable extension or shortening of shaft length, a thrust bearing near the motor end of the machine will permit shaft to move laterally at the free end.

Tapered roller bearings give positive shaft alignment and carry high thrust loads. Adjustable mountings are required to take up and limit end play.

Worm shafts in which the direction of rotation reverses should be provided with thrust bearings of sufficient capacity to absorb thrust loads in both directions.

Adjustable outboard bearings on the ends of long worm shaft extensions from motors prevent distortion and consequent faulty meshing.

LUBRICATION. Breakaway torque, that is, torque required to overcome starting friction caused by cold lubricating oil, poor distribution of lubricant, and dry spots in bearings, varies with the type and amount of lubricant, temperature, and frequency of starting.

Viscosity of lubricant should be selected on the basis of ambient and running temperatures. Shot lubrication before starting machines eases starting loads on motor and reduces wear of rubbing surfaces.

Oil baffles, slingers, and labyrinth packings on the inboard side of bearings in inclosed machines reduce the loss of oil caused by circulation of air through the bearing. In dusty locations, these devices help prevent abrasive dust reaching shaft and bearings.

Oil-saturated woven felt washers can be used to lubricate ball bearings.

Passages for return of lubricant to the supply reservoir should be large enough to provide ample circulation. Oil wells should be large and have an oil filler opening on top and a drain plug at low point.

Overgreasing and subsequent overheating of bearings can be prevented with pressure relief arrangements on grease chambers.

Bearings operating in high temperature can be lubricated by oil bleeding into them from grease in an adjoining cavity. This method of lubricating bearings prevents sticking caused by soap base remaining in the bearings after a high temperature operation.

Torque Required to Ac

To accelerate any part of a machine from rest to a given velocity, energy must be given to the part. This energy, neglecting friction and windage losses, is equal to the energy possessed by the part when moving at the given velocity. Where

W = weight of part, lb
V = linear velocity, ft per min
g = acceleration of gravity, ft per sec per sec
N = revolutions, per min
R = radius of gyration of part, ft
E = energy, ft-lb

Kinetic energy in a sliding part that moves in a straight line, such as carriages, tool heads and pistons, is

$$E = \frac{W}{2g}\left(\frac{V}{60}\right)^2 = \frac{WV^2}{231{,}840}, \text{ ft-lb} \quad (7)$$

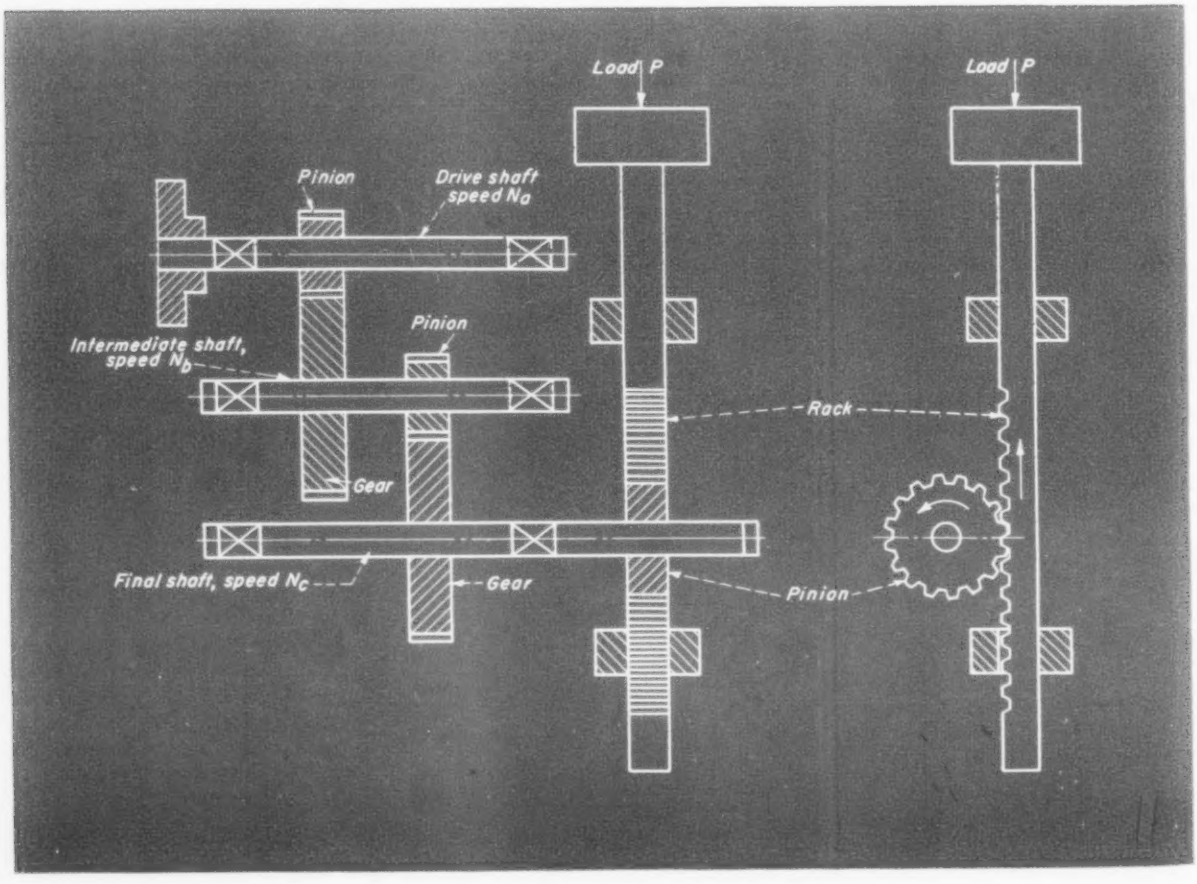
Kinetic energy in parts that rotate about an axis, such as gears, pulleys and rotors, is

$$E = \frac{W}{2g}\left[2\pi R\left(\frac{N}{60}\right)\right] = \frac{WR^2N^2}{5{,}872}, \text{ ft-lb} \quad (8)$$

This equation is derived from Eq (7) by substituting for the linear velocity V the velocity of the point in the rotating part, at distance R from the axis, at which all the weight of the part can be considered to be concentrated and the moment of inertia of the mass of the part remain the same, thus

$$V = 2\pi R N, \text{ ft per min}$$

Force required to accelerate a part moving in a straight line, and the torque required to accelerate a rotat-



to Accelerate Machines

ing part, is always greater than that necessary to maintain motion at a constant velocity.

The force F required to accelerate a part from rest to a velocity V (ft per min) in a straight line in t seconds is

$$F = \frac{W(V)}{g(60t)} = \frac{WV}{1,932\,t}, \text{ lb}$$

or

$$WV = 1,932\,tF$$

or

$$WV^2 = 1,932\,tFV \quad (9)$$

The torque T required to accelerate a part from rest to a rotation of N revolutions per minute about its axis in t seconds is

$$T = FR = \frac{W}{g}\left(\frac{2\pi RN}{60\,t}\right)R$$

$$= \frac{WR^2\,N}{308\,t}, \text{ lb-ft} \quad (10)$$

$$\text{or } WR^2 = \frac{308\,tT}{N} \quad (11)$$

Normally in the power train of a machine the rotating parts, such as gears and drums, rotate at a different speed than that of the driving motor. To calculate the torque required to accelerate such a system, the WR^2 of each moving part is reduced to an equivalent WR_e^2 about a common shaft, usually the motor shaft, and then the WR_e^2 of the respective parts are added together.

The equivalent WR_e^2 of a rotating part referred to the motor shaft is obtained by means of the energy Eq (8), thus where N_m is the rpm of the motor shaft the kinetic energy in the rotating part is

$$E = \frac{WR^2\,N^2}{5,872} = \frac{WR^2_e\,N^2_m}{5,872}$$

or

$$WR^2_e = WR^2\,(N/N_m)^2 \quad (12)$$

Referred to the motor shaft, the equivalent WR^2_e of a part moving in a straight line is obtained from Eqs (7) and (8) in the same manner, thus

$$E = \frac{WV^2}{231,840} = \frac{WR^2_e\,N^2_m}{5,872}$$

or

$$WR^2_e = \frac{WV^2}{39.48\,N^2_m} \quad (13)$$

Example: Compute the total equivalent WR^2_e at the drive shaft for the power train shown here.

Where

V = lineal velocity of rack and load, ft per min
A = total WR^2 of rotating parts on the drive shaft
B = total WR^2 of rotating parts on intermediate shaft
C = total WR^2 of rotating parts on final shaft
D = WV^2 of lineally moving rack and load
N_a = rpm of drive shaft
N_b = rpm of intermediate shaft
N_c = rpm of final shaft
WR^2_e = total equivalent WR^2_e referred to drive shaft for complete power train

$$WR^2_e = A + B\,(N_b/N_a)^2 + C\,(N_c/N_a)^2 + D/39.48N^2_a \quad (14)$$

Torque required at the drive shaft to accelerate this WR^2_e load from rest to a drive shaft speed of N_a rpm in t sec from Eq(10) is

$$T_a = \frac{WR^2_e\,N_a}{308\,t}, \text{ lb-ft}$$

In a machine where the motor is coupled directly to the drive shaft, the T_m required at the motor shaft to accelerate the WR^2_e of the load and the WR^2_m of the motor is

$$T_m = \frac{(WR^2_e + WR^2_m)\,N_a}{308\,t}$$

When the motor is connected to the drive shaft of the machine by a belt, the motor having a speed of N_m rpm and the drive shaft a speed of N_a rpm, the torque T_m required at the motor shaft to accelerate the WR^2_e of the load referred to the drive shaft and the WR^2_m of the motor referred to its shaft is

$$T_m = \frac{WR^2_m\,N_m + WR^2_e\,(N_a/N_m)^2}{308\,t}$$

In addition to the torque required for acceleration, additional torque is required to overcome load torque, friction torques and break-away torques.

Torque at the drive shaft caused by load P from Eq (5) is

$$T'_a = \frac{T_c N_c}{N_a}, \text{ lb-ft}$$

or at the motor shaft

$$T'_m = T'_a\,N_a/N_m$$

Torque that results from the friction on parts such as bearings, gears and windage, and break-away torque caused by the static friction of oil and grease in bearings, are often large enough to be considered. These torques can be expressed in terms of equivalent torque T''_m at the motor shaft by using the same procedure as with other torques.

To start and accelerate a machine and to overcome load and friction torques, therefore, the motor must produce a torque T on its shaft such that

$$T = T_m + T'_m + T''_m$$

WR^2 of Parts

WR^2 of rotating parts varies directly as the weight of the unit volume of the material used. For a part of given dimensions, therefore, a smaller WR^2 can be obtained by using a material having a smaller weight per unit volume.

WR^2 of rotating parts can sometimes be reduced by decreasing the diameter of the part.

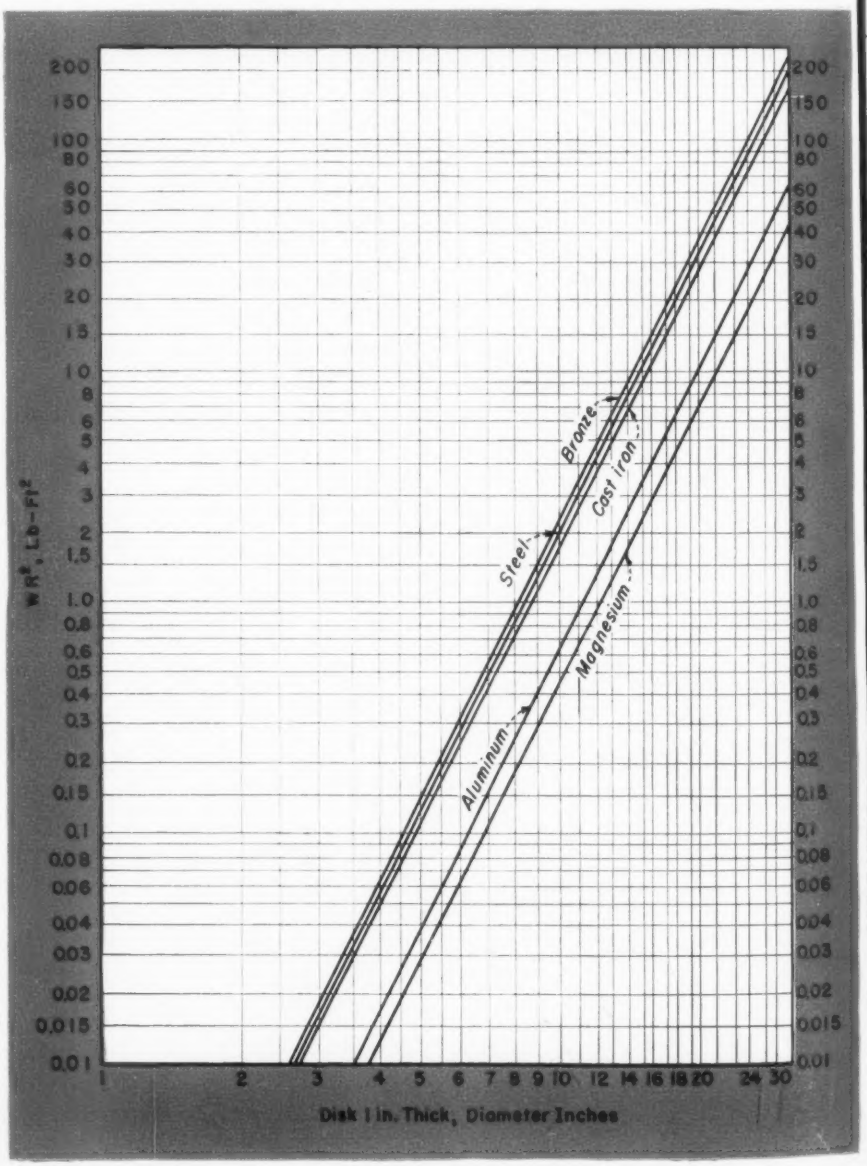
WEIGHT OF METALS,

Pounds per Cubic Inch

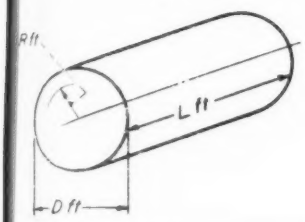
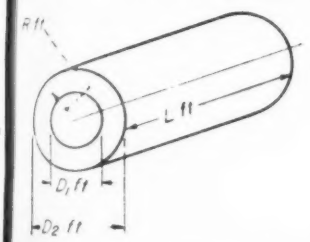
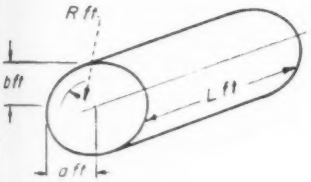
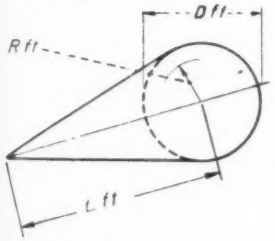
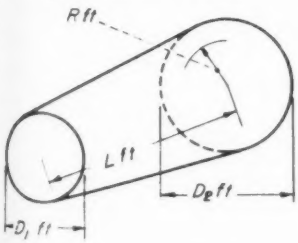
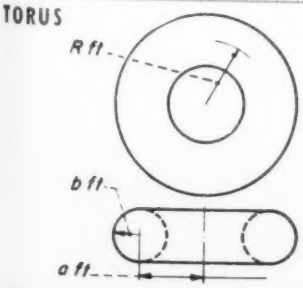
Magnesium	0.0628
Aluminum	0.0924
Cast iron	0.260
Steel	0.282
Copper	0.318
Bronze	0.320
Lead	0.411

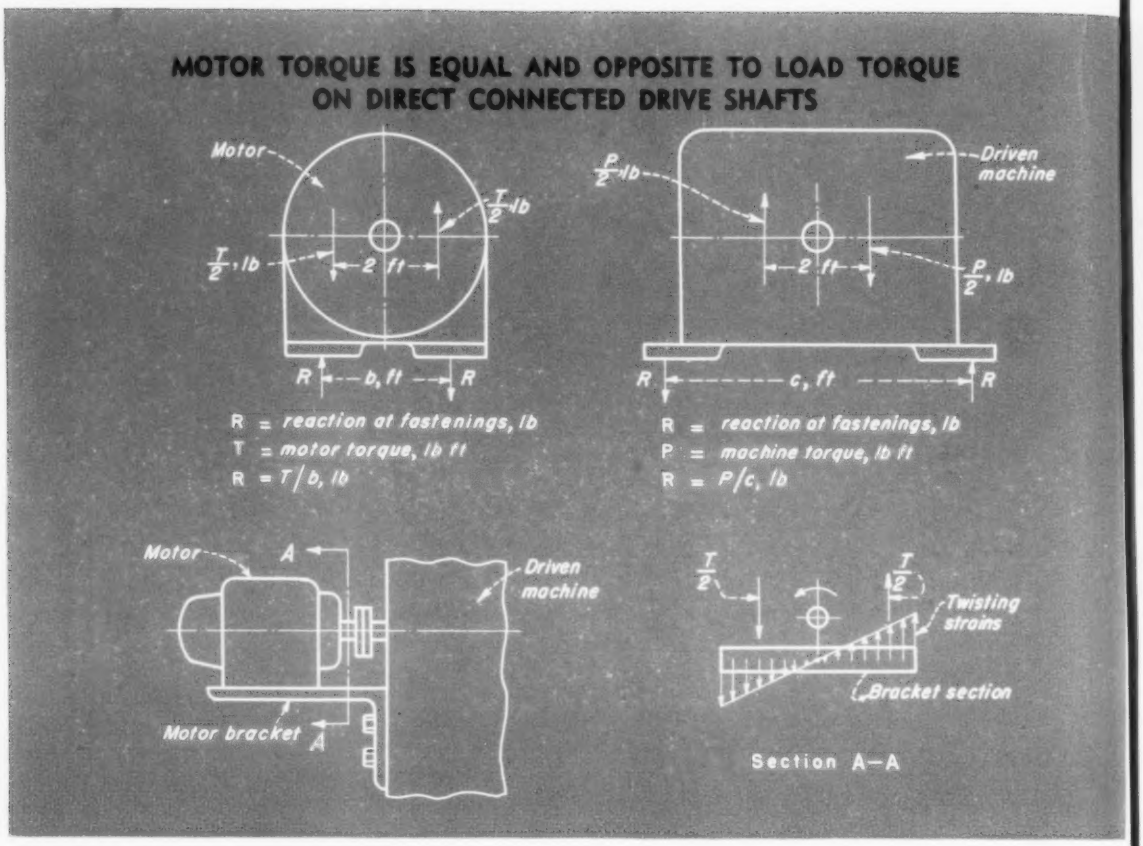
Parts having complex shapes of revolution often can be considered to consist of component or elemental disks. For such shapes, the WR^2 of the part can be readily calculated by adding together the WR^2 of the elemental disks that make up the part. The log-log chart presented here gives the WR^2 of 1 in. thick disks for a range of diameters and different metals.

On this chart, diameters are plotted in inches and WR^2 in lb-ft squared.



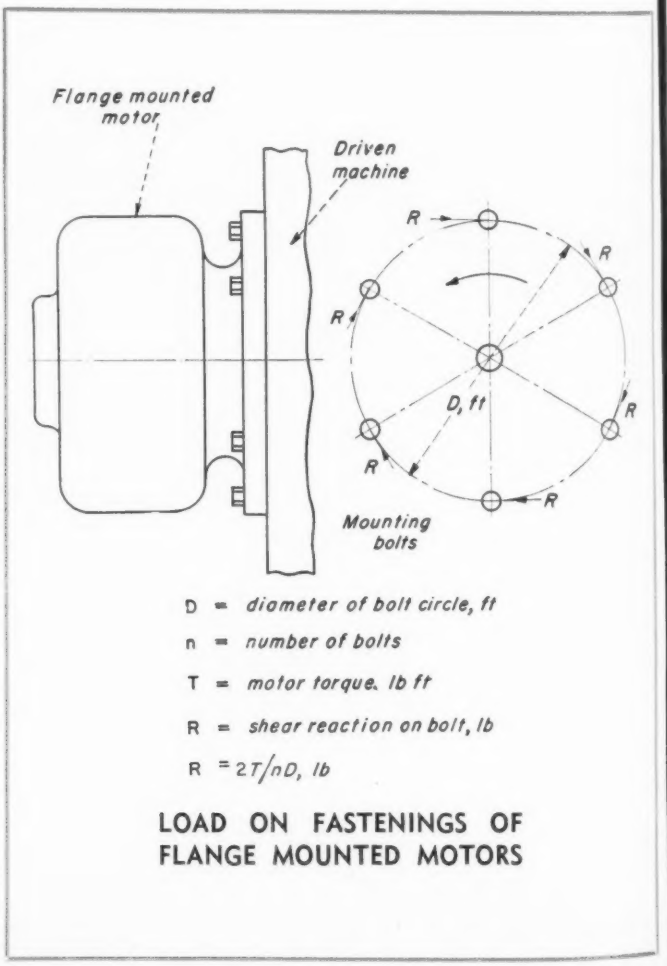
WR² OF PARTS

Part	Radius of Gyration R, feet	Weight W, lb w = Weight per cu in. of material, lb	WR,² lb-ft²
CIRCULAR CYLINDER 	0.354 D	1.360 w L D²	170.4 w L D⁴
HOLLOW CIRCULAR CYLINDER 	$0.354\sqrt{D_2^2 + D_1^2}$	$1{,}360\ w\ L\ (D_2^2 - D_1^2)$	$170.4\ w\ L\ (D_2^4 - D_1^4)$
ELLIPTICAL CYLINDER 	$\sqrt{\frac{a^2 + b^2}{2}}$	5,426 a b w L	$5{,}426\ a\ b\ w\ L\left(\frac{a^2 + b^2}{2}\right)$
CIRCULAR CONE 	$\frac{D}{3.66}$	452 w D² L	33.7 w D⁴ L
FRUSTUM OF CONE 	$0.275\sqrt{\frac{D_2^5 - D_1^5}{D_2^3 - D_1^3}}$	$[453\ (D_2^2 + D_1^2) + 511\sqrt{D_2^2\ D_1^2}]\ w\ L$	$\left(\frac{D_2^5 - D_1^5}{D_2^3 - D_1^3}\right)\left[34\ (D_2^2 + D_1^2) + 38.3\sqrt{D_2^2\ D_1^2}\right] w\ L$
TORUS 	$0.5\sqrt{4\ a^2 + 3\ b^2}$	33,938 w a b²	8,485 w a b² (4 a² + 3 b²)



Bedplates, Mounting Brackets and Flanges

Bedplates, bracket structures, and flanged arrangements for mounting motors to driven machines should be designed not only to permit easy assembly and maintenance, but also stiff enough in torsion to resist the torque of the motor.



Speed of Motor

Selection of motor speed is an important factor that affects the performance of a motor.

A 2 hp two-speed motor (1800/900 rpm, single winding, open frame, continuous duty, frame size 225) was used to drive a grinder feed. This motor became too hot and the windings burned out. The mechanical designer decided the motor was deficient in horsepower. Since space was limited, the designer's solution was to substitute a 3600/1800 rpm, two-speed 3 hp motor using the same frame. With the higher speed motor, burnouts were worse than before.

INVESTIGATION REVEALED

1. Motor was stopped and reversed by plugging at rates as high as 25 times per min. Also on some cycles the motor speed was changed as frequently as 10 to 15 times per minute.
2. Torque required for operation was about 0.7 lb-ft. This torque was essentially constant at all speeds.

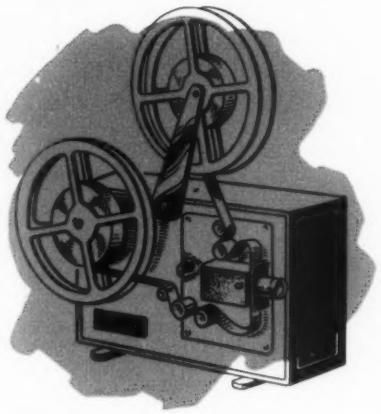
Substituting a motor having the same size frame and twice the speed was not a correct solution, because the higher speed motor has a lower torque speed curve, that is, the area under the curve is smaller. The acceleration torque delivered by the motor, therefore, will be less.

When a motor is plugged from one direction to the reverse, the heat generated in the motor is approximately four times that generated in starting. The heat generated depends upon a motor design ratio factor.

It was evident that high temperature resulted not from running load, but from inrush current at starting, plugging and speed change. Since a motor frame produces constant torque, the frame 225 will produce practically the same torque at lower speeds.

The losses resulting from the inrush current could be calculated, but the cycles were so different that the answer would be indefinite. The practical solution was to build a 900/450 rpm motor and rate it $\frac{3}{4}$ hp, and to make suitable pulley and belt changes to obtain correct final speed. This motor produced $4\frac{1}{4}$ lb-ft torque at the high speed. Test proved this motor satisfactory.

Later tests proved that a frame 204 gave about $\frac{1}{4}$ hp, or about $1\frac{1}{2}$ lb-ft torque at 900 rpm, which was ample. Lower speeds lowered inrush current and also resulted in a lower WR^2 to be accelerated.



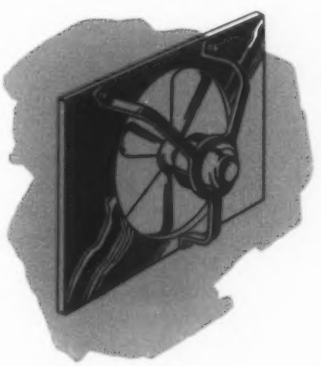
Case History. In a portable motion picture projector driven by a universal motor through a low ratio belt drive, a manufacturer desired to have a relatively low operating speed. The circuit included a rheostat and high starting torque was needed.

These two requirements could not be met with an efficient motor having the necessary low speed rating.

Increasing the pulley ratio between the motor and the projector, permitted the use of a moderate speed motor that could produce a high starting torque at reduced speed.

Case History. To drive an evaporator fan at about 1750 rpm, a manufacturer wanted a motor that would operate on both alternating current and direct current. To build an efficient universal motor for this speed is difficult.

The solution was to change the size and design of fan so that it could be driven at about 3,000 rpm. At this speed a standard universal motor could be used, thus the customer got a less expensive motor.



Load and Thermal Capacity

Motors should not be selected solely on the basis of torque and power required at maximum load, except when it is known from experience that the characteristics of a given motor will give satisfactory performance.

Torque and power requirements of the duty cycle should be carefully analyzed before a size and type of motor is selected, especially when starting, stopping, and reversing of load are frequent. Torque and power requirements should be measured by test or computed for periods of full load, partial load, overload, starting and reversing.

Standards for motor temperature ratings define safe limits of operation only under definite conditions. Such ratings do not take into consideration all the conditions met with in actual service.

With the duty cycle, torque and power requirements determined, a motor should be selected that has the necessary thermal capacity and operating characteristics for proper performance.

Excessive overloading may cause a breakdown, but appreciable underloading is uneconomic when motor capacity is not used.

Mechanical strength of shafts, bearings and couplings; and effectiveness of lubrication; are factors that limit the amount of safe overloading.

Overloading should also be considered in relation to the temperature-rises permitted by the motor rating. Overloading is limited by the temperature reached under the most severe ambient temperature conditions.

Effect of temperature is important since it controls the life of the insulation. Electrical and mechanical properties deteriorate at all operating temperatures. Rate of deterioration depends upon both temperature and time.

Heat produced in a motor varies as the square of the current, therefore, successive and frequent starting, reversing or momentary overloads, which take more than normal running current, may produce a temperature rise exceeding that permitted.

Uneconomic utilization of motor capacity frequently results when a horsepower rating is selected on the basis of a peak loading that occurs for a comparatively short time during an intermittent load or a varying load. The thermal capacity of a motor is not used adequately under such loading if the temperatures reached are below the maximum permissible.

Where there is a reserve of thermal capacity during short periods of the duty cycle, it is generally permissible to operate with a reasonable overload during a period of peak loading.

Permissible overloads for Standard 60 cycle polyphase induction motors for different ambient temperatures and hours of use are given in the accompanying Load-Time-Temperature Table.

Excess motor capacity can be checked by investigating the loading over representative duty cycles and times. With known or calculated values of load and time for a duty cycle, the thermal effect of the different loadings are determined so as to ensure that temperature conditions over the representative period are not less favorable than those which obtain when the motor is operating at its continuous rating.

Where

t_o = units of time motor is stopped during duty cycle
K = cooling factor
H_1, H_2, H_3 = hp for t_1, t_2, t_3 units of time respectively motor is running during duty cycle
H = root mean square hp, that is, equivalent continuous load for time interval $t_o + t_1 + t_2 + t_3$

the varying load can be resolved into an equivalent continuous load from

$$H = \sqrt{\frac{H_1^2 t_1 + H_2^2 t_2 + H_3^2 t_3}{K t_o + t_1 + t_2 + t_3}} \qquad (1)$$

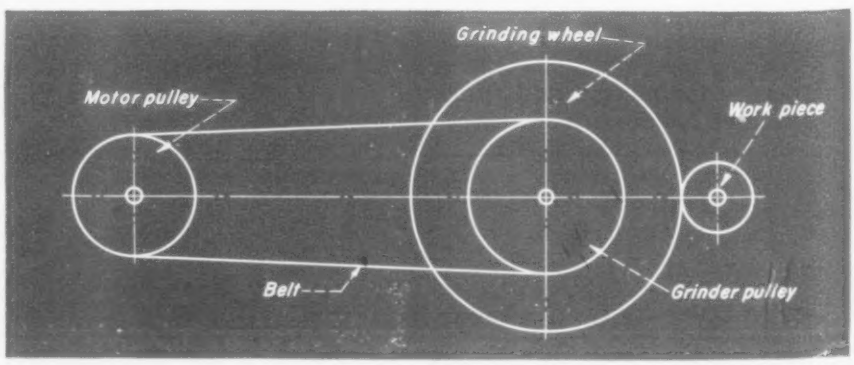
Cooling effect is reduced when the motor is at rest, therefore, if the motor is stopped for part of the cycle, a value of K equal to $\frac{1}{3}$ should be used for the rest period in determining the effective cycle time for open motors; or for inclosed motors K equal to $\frac{1}{2}$ should be used.

For fast repeating cycles that include reversals and deceleration by plugging, the additional heating caused by reversing and external WR^2 loads must be taken into consideration, and a more elaborate duty cycle analysis is required.

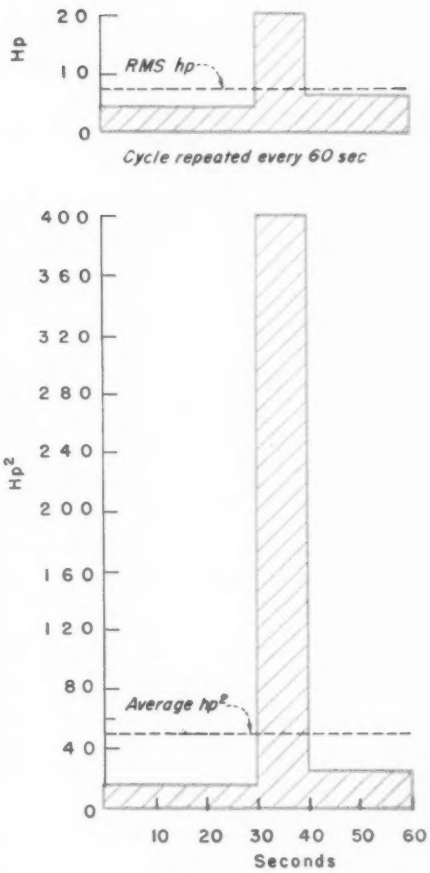
In addition to determining the horsepower rating with reference to heat capacity, a check should also be made to assure that the peak torques, even though of short duration, imposed by the driven machine do not exceed the breakdown torque the motor will develop without stalling. The breakdown torque value of the motor should always exceed the peak torque imposed by the duty cycle. Preferably, the breakdown torque of the motor should be at least 25 percent greater to compensate for unusual conditions, such as low voltage and high friction.

Using this method it is often practicable to overload a motor for an appreciable time, provided that the loading at other times in the duty cycle of the machine is sufficiently low to avoid excessive temperature rise.

EXAMPLE. A grinding machine with a 12 in. dia wheel is driven by a Design B motor through a belt. Full load motor speed is 1,750 rpm. Grinding wheel speed is 1,580 rpm. Work cycle is repeated every 60 sec. Work cycle consists of: Machine idles for 30 sec when load is 4 hp, first cut requires 5 sec when load is 20 hp, and sizing cut requires 25 sec when load



# of Induction Motors



Duty cycle of grinding machine as specified in the example. Upper graph is plotted in units of horsepower. Lower graph is plotted in units of horsepower squared.

is 5 hp. The work cycle is shown in the upper part of the accompanying graph.

A motor having a rating of 20 hp to meet the peak demand will be underloaded for most of the cycle. From thermal considerations, the rating should be such that the total heating at continuous rating is equal to the total heating with intermittent loading.

The continuous rating for equal heating from Eq (1) is

$$H = \sqrt{\frac{4^2 \times 30 + 20^2 \times 5 + 5^2 \times 25}{30 + 5 + 25}} = 7.2$$

In the lower part of the graph, the ordinates are horsepower squared and the average horsepower squared. The RMS horsepower is shown in the upper part of the graph. From thermal considerations, a motor of 7½ hp continuous rating for the duty cycle shown would be satisfactory. But this would mean that a 7½ hp motor would be required to meet a maximum torque needed for a peak of 20 hp, that is, the peak torque would be 266 percent of rated load running torque.

A torque of 266 percent of rated load running torque is in excess of the 215 percent breakdown torque produced by this motor, and would probably cause excessive mechanical stresses and also stall the motor.

A continuously rated motor of 10 hp would adequately meet the load cycle and the torque demand at 20 hp, since the breakdown torque rating of a Design B, 10 hp motor is 200 percent of full load torque.

In this grinder application, the motor cannot have an open frame, because dust and grit would get into the windings. Nor can the motor be fan cooled, because dust and grit would collect on the fan and quickly unbalance it. The optimum solution, even though the frame size is larger and cost is greater, is to use a fully inclosed, non-ventilated motor rather than an open frame motor.

Continuous duty rating of motors that drive hand fed machine tools should be selected on the basis of the thermal capacity necessary to avoid overheating during their expected or measured maximum work cycle.

Load—Time—Temperature Table For Motors

Standard 60-cycle polyphase induction motors with speeds of 450 to 3600 rpm, hp ratings of 1 to 200, and Class A insulation.

Max allowable occasional ambient* temp C when motor is fully loaded during the following number of hours per day			Permissible load in percent of motor nameplate rating	Approx motor-temp rise C by thermometer	Allowable momentary peak load in percent of permissible load, assuming 10 percent voltage reduction
24 hr	8 hr	2 hr			
Open Motors, Rated 40 C					
40	50	60	115	50	140
50	60	70	100	40	160
60	70	75	80	30	200
Splashproof Motors, Rated 50 C					
40	50	60	100	50	160
50	60	70	85	40	180
60	70	75	70	30	220
Totally Inclosed; and Totally Inclosed, Fan-Cooled Motors, Rated 55 C					
40	50	60	100	55	160
50	60	70	85	45	188
60	70	75	70	35	220

* The usual ambient temperatures should be 5 to 10 C lower than the maximum allowable occasional ambient temperature listed here.

Polyphase Squirrel-Cage Induction Motors

Speed-Torque Characteristics

Torque, horsepower and speed requirements demanded in drives for most machines can be met with one of five designs of squirrel cage polyphase induction motors. These five designs are described in NEMA Standards MG50-119, MG9-523, MG9-535, and MG9-538. These standards were adopted August 7, 1947 and became effective January 1, 1948. Each design offers a different combination of torque, speed and current characteristics to meet the operating requirements of various industrial applications.

All five designs can withstand full voltage starting directly across the power lines, that is, the motors are strong enough mechanically to withstand magnetic stresses and the locked-rotor torques developed at the time the switch is closed.

DESIGN A produces exceptionally high breakdown torques but at the expense of high locked rotor currents that normally require provision for starting with reduced voltage. This motor is suitable for machines in which the friction and inertia loads are small.

DESIGN B has normal starting torque adequate for a wide variety of industial machine drives and a starting current usually acceptable on power systems. This design is suitable where slightly more than full load torque and low slip is required, also where relatively high breakdown torque is needed to sustain occasional emergency overloads, or where a low locked-rotor current is needed. These motors are for use in driving machine tools, blowers, centrifugal pumps, and textile machines.

DESIGN C has high starting torque and a normal breakdown torque. Applications for this design are machines in which inertia loads are high at starting, but normally run at rated full load and not subjected to high overload demands after running speed has been reached. Conveyors, plunger pumps, compressors that are not unloaded at starting and oven chain conveyors; also hoists, cranes and machine tools where a quick start and reversal are required; are typical examples of such machines.

DESIGN D develops extremely high starting torque with moderate starting current. This design uses a high resistance type rotor to obtain variation of speed with load and has no sharply defined breakdown torque. This motor eases off in speed when surge loads are encountered and also develops high torque to recover speed rapidly. Typical applications for this motor are machines in which heavy loads are suddenly applied and removed at frequent intervals, such as hoists, machines with large flywheels, conventional punch presses, and centrifuges.

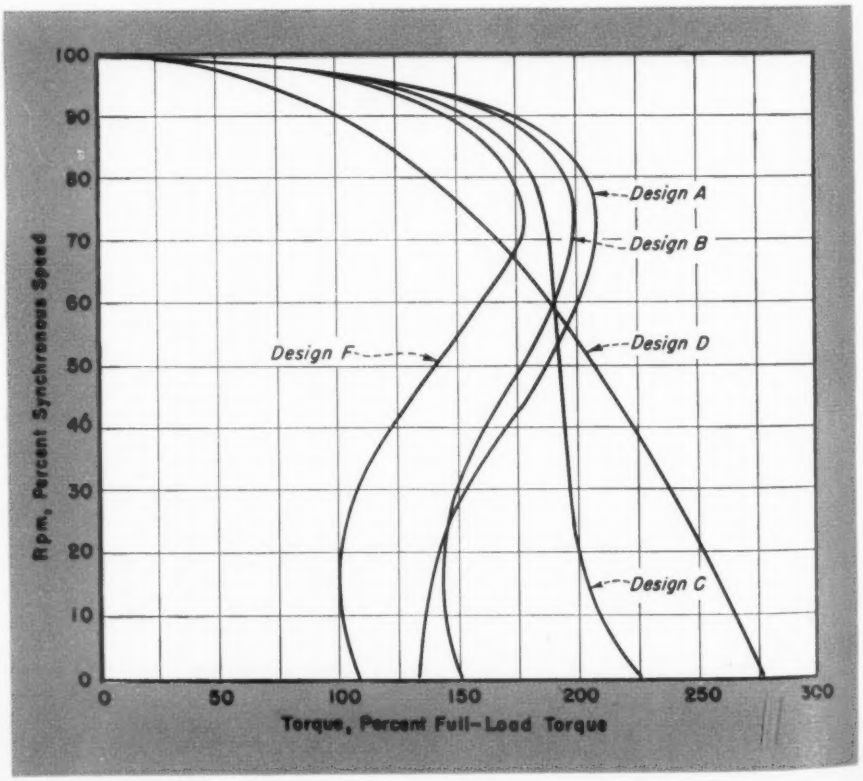
DESIGN F is built to obtain a low locked rotor condition. In this motor, the starting torque and breakdown torque are both low, therefore, the field of application is small. Typical applications for which this motor is well suited, but seldom encountered, are drives where the torque required to start and accelerate light weight rotating parts is small such as high speed cutter heads in woodworking machines.

SLIP RATINGS of the five designs are:

Design	Slip, percent
A	less than 5*
B	less than 5*
C	less than 5
D	5 or more
F	less than 5

* Motors with 10 and more poles may have slip slightly greater than 5 percent.

LOCKED-ROTOR TORQUE. Specific values of locked-rotor torque developed by Designs A, B, and C motors are given in the accompanying tables.



Locked-Rotor Torque of Designs A and B Motors With Continuous Ratings

Values are expressed in percent of full-load torque, and represent the upper limit of the range of application.
Values are based on rated voltage and frequency.

Hp	Frequency	Synchronous Speeds, rpm							
	60 cycles 50 cycles	3600 3000	1800 1500	1200 1000	900 750	720 600	600 500	514 428	450 375
½					150	150	115	110	105
¾				175	150	150	115	110	105
1			275	175	150	150	115	110	105
1½		175	265	175	150	150	115	110	105
2		175	250	175	150	145	115	110	105
3		175	250	175	150	135	115	110	105
5		150	185	160	130	130	115	110	105
7½		150	175	150	125	120	115	110	105
10		150	175	150	125	120	115	110	105
15		150	165	140	125	120	115	110	105
20		150	150	135	125	120	115	110	105
25		150	150	135	125	120	115	110	105
30		150	150	135	125	120	115	110	105
40		135	150	135	125	120	115	110	105
50		125	150	135	125	120	115	110	105

Locked-Rotor Torque of Design C Motors with Continuous Ratings

Values are expressed in percent of full-load torque, and represent the upper limit of the range of application.
Values are based on rated voltage and frequency.

Hp	Frequency	Synchronous Speeds, rpm		
	60 cycles 50 cycles	1800 1500	1200 1000	900 750
3			250	225
5		250	250	225
7½		250	225	200
10		250	225	200
15		225	200	200
20		200	200	200
25 and larger		200	200	200

Locked-rotor torques of Design D, 60 and 50 cycle, 4, 6, and 8 pole motors, with rated voltage and frequency applied, is 275 percent of full-load torque. This value represents the upper limit of application.

Locked-rotor torques of Design F, 60 and 50 cycle, 4 and 6 pole motors, rated 30 hp and larger, with rated voltage and frequency applied, is 125 percent of full-load torque, which represents the upper limit of application.

BREAKDOWN TORQUES of Designs B and C motors are given in the accompanying Table. Breakdown torques of Design A motors are in excess of the values given for Design B motors. Breakdown torque of Design F motors, 30 hp and larger, at all speeds is 135 percent of full-load torque.

It is usually good practice to apply motors at momentary loads at least 20 percent below the values given for maximum torque in order to offset that much torque drop caused by an allowable 10 percent voltage drop.

Breakdown Torque of Designs B and C, 60 and 50 Cycle Motors With Continuous Ratings

Values are expressed in percent of full-load torque, and represent the upper limit of the range of application.
Values are based on rated voltage and frequency.

Hp	Synchronous Speed, rpm	Design B	Design C
½	900–750	250	
	lower than 750	200	
¾	1200–1000	275	
	900–750	250	
	lower than 750	200	
1	1800–1500	300	
	1200–1000	275	
	900–750	250	
	lower than 750	200	
1½	3600–3000	300	
	1800–1500	300	
	1200–1000	275	
	900–750	250	
	lower than 750	200	
2	3600–3000	275	
	1800–1500	275	
	1200–1000	250	
	900–750	225	
	lower than 750	200	
3	3600–3000	250	
	1800–1500	275	
3	1200–1000	250	225
	900–750	225	200
	lower than 750	200	
5	3600–3000	225	
	1800–1500	225	200
	1200–1000	225	200
	900–750	225	200
	lower than 750	200	
7½	3600–3000	215	
	1800–1500	215	190
	1200–1000	215	190
	900–750	215	190
	lower than 750	200	
10	3600–3000	200	
	1800–1500	200	190
	1200–1000	200	190
	900–750	200	190
	lower than 750	200	
15 and larger	all speeds	200	190

Multispeed Machine Drives

Two, three or four different machine speeds can be economically obtained with multispeed polyphase squirrel-cage induction motors. These motors may have electrical characteristics corresponding approximately to those of any one of the five designs of the integral horsepower polyphase squirrel cage induction motors. Similar characteristics are obtained at all speeds.

Two-speed squirrel cage motors have either a single winding or two windings in the stator.

Single-winding two-speed motors are usually built in the frame size corresponding to the horsepower and speed of the lower speed rating. Combinations of synchronous speeds in rpm for two-speed, single-winding 60 cycle motors are usually limited to

3,600/1,800, 1,800/900,
1,200/600, 900/450

Two-winding two-speed motors are also built for these speed combinations. Two windings are always required if the desired speeds are not in a 2 to 1 ratio.

Three-speed, three-phase, three-winding motors are not recommended by most manufacturers because too much space is needed for winding clearances and insulation. Size of frames required for three-winding motors are larger than for two-winding motors.

Most three-speed motors are built with two windings. The two speeds produced by one winding must have a ratio of 2 to 1, while the third speed need have no relation to the first two. Synchronous speeds of three-speed two-winding 60 cycle motors are usually limited to the same combinations available for two-speed, single-winding motors plus one other speed of any commercial value.

Four-speed 60 cycle motors are usually limited to the following synchronous rpm speeds

1,800/1,200/900/600
1,200/900/600/450

Multispeed polyphase squirrel cage motors are built in three types:

1. Constant-torque motors
2. Constant-horsepower motors
3. Variable-torque motors

CONSTANT-TORQUE MOTORS produce practically the same starting and maximum torques at the several speeds. Since horsepower output is proportional to the product of torque and speed, the horsepower output varies directly as the speed. For example, a two-speed constant-torque 1,800/900 rpm motor rated 20 hp at 1,800 rpm would be rated 10 hp at 900 rpm.

Constant-torque motors are used where the torque required by the drive is approximately the same regardless of speed, such as in conveyors, stokers, reciprocating compressors, and printing presses.

CONSTANT-HORSEPOWER MOTORS have the same horsepower rating at all speeds. These motors, therefore, produce starting, maximum and full load torques that are inversely proportional to the speeds. For example, a 1,800/900 rpm motor rated at 20 hp at 1,800 rpm would also be rated 20 hp at 900 rpm.

Constant-horsepower motors are used to drive machines in which the torque required is approximately inversely proportional to the speed. For instance, lathes are frequently designed to take a heavy roughing cut at a low speed and a light finishing cut at a high speed. In this application, the constant-horsepower motor delivers a high torque at low speed and a low torque at high speed. Other applications are drives for boring mills, and radial drills.

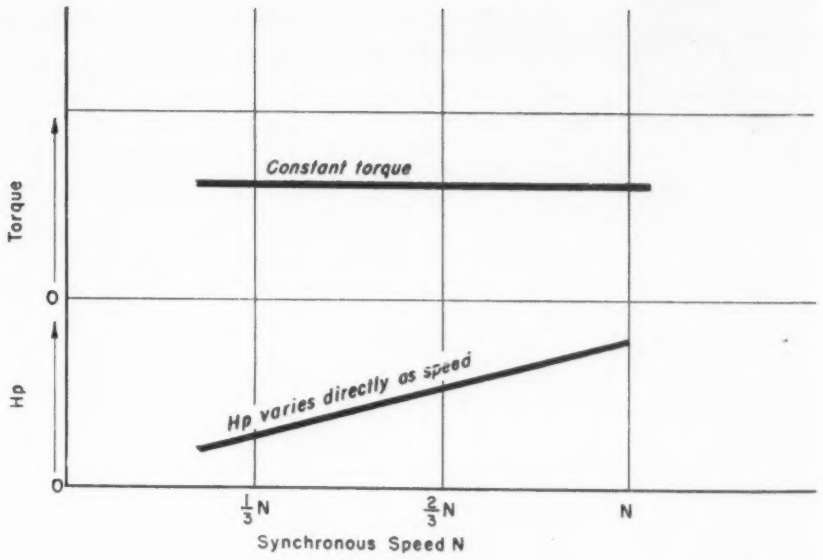
VARIABLE-TORQUE MOTORS produce starting and maximum torques that are proportional to the speed. The horsepower delivered at the several speeds, therefore, varies as the square of the speed. For example, a two-speed variable torque 1,800/900 rpm motor rated 20 hp at 1,800 rpm would be rated 5 hp at 900 rpm.

Variable-torque motors are used to drive machines such as centrifugal pumps, fans, blowers (not including positive pressure blowers), and other machines in which the horsepower required decreases as rapidly as the square of the speed.

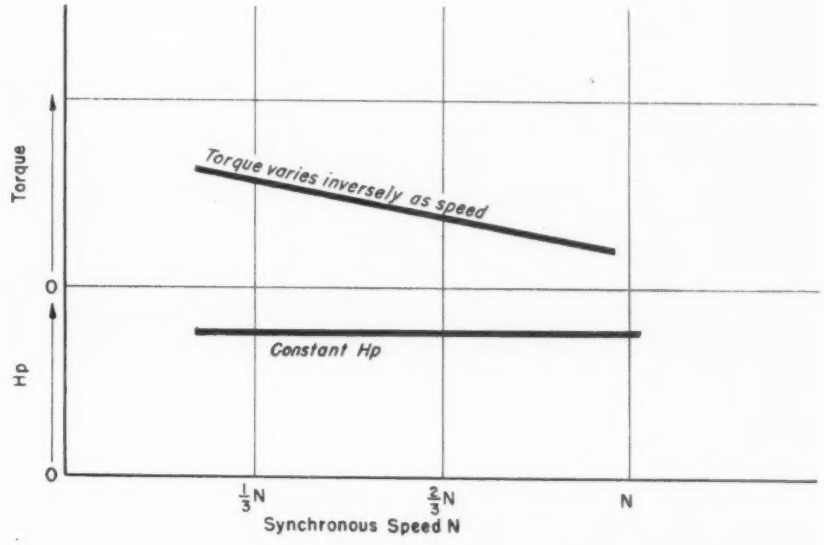
Typical Applications For Multispeed Motors

Constant Torque	Constant Horsepower	Variable Torque
Breadmixers	Lathes	Ventilating fans
Drill heads	Some punch presses	Blowers
Machine tools		Unit heaters
Positive pressure blowers		Centrifugal pumps
Elevators		
Laundry machinery		
Stokers		
Compressors		
Conveyors		
Extractors		

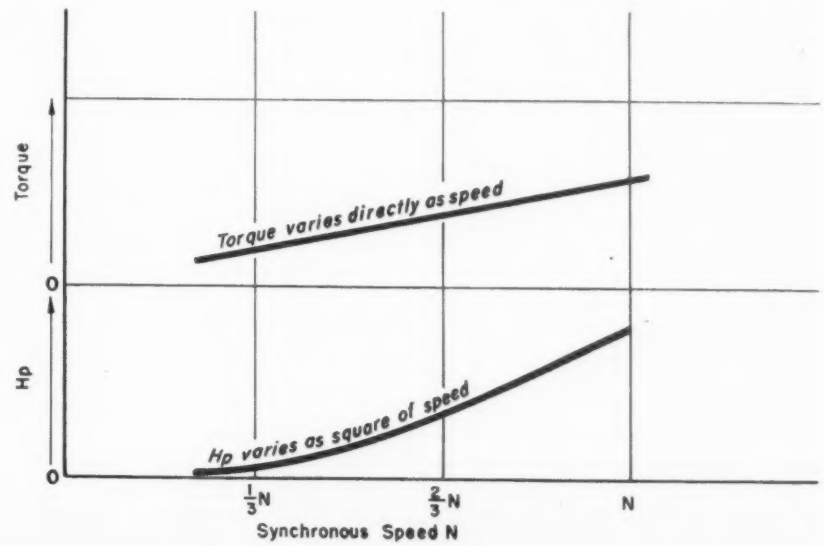
Constant Torque Motor

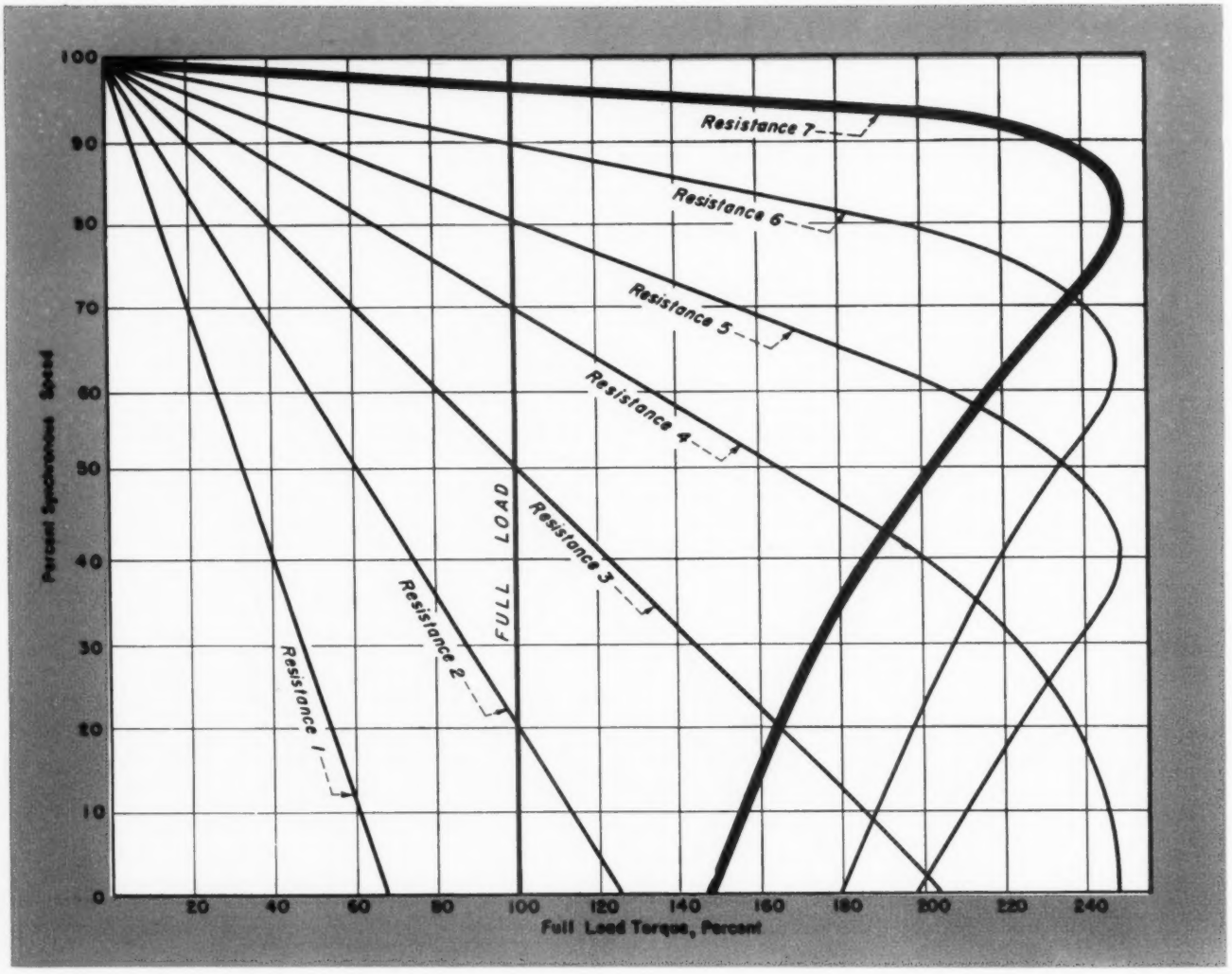


Constant Hp Motor



Variable Torque Motor





Speed-Torque Characteristics of Polyphase Wound-Rotor Motors

Characteristics of wound-rotor motors are such that the slip depends almost entirely upon the load on the motor. The speed returns practically to maximum when the load is removed. This characteristic limits the use of these motors on applications where reduced speeds at light loads are desired.

Wound-rotor motors are suitable for constant-speed applications that require frequent starting or reversing under load, or where starting duty is severe and exceptionally high starting torque is required.

Second major type of application is where speed adjustment is required. Controllers are used with these motors to obtain adjustable speed over a considerable range. But at any point of adjustment the speed will vary with a change in load. It is usually not practicable to operate at less than 50 percent of full speed by introducing external resistance in the secondary circuit of the motor.

Horsepower output at 50 percent of normal speed is approximately 40 percent of rated horsepower.

Use of wound-rotor induction motors has been largely in continuous duty constant speed applications where particularly high starting torques and low starting currents are required simultaneously, such as in reciprocating pumps and compressors. These motors are also used where only alternating current is available to drive machines that require speed adjustment, such as some types of fans and conveyors.

Typical torque-speed characteristics of wound-rotor motors are shown in the accompanying curves for full voltage and for reduced voltages that are obtained with different values of secondary resistance.

Table I—Range of Breakdown Torques For Defining Horsepower

Revolutions Per Minute						
3600 3450	3000 2850	1800 1725	1500 1425	1200 1140	900 850	Synchronous rpm Approx F L rpm
Breakdown Torque, oz-ft						Brake Hp Rating
2.0– 3.7	2.4– 4.4	4.0– 7.1	4.8– 8.5	6.0–10.4	8.0–13.5	1/20
3.7– 6.0	4.4– 7.2	7.1–11.5	8.5–13.8	10.4–16.5	13.5–21.5	1/12
6.0– 8.7	7.2–10.5	11.5–16.5	13.8–19.8	16.5–24.1	21.5–31.5	1/8
8.7–11.5	10.5–13.8	16.5–21.5	19.8–25.8	24.1–31.5	31.5–40.5	1/6
11.5–16.5	13.8–19.8	21.5–31.5	25.8–37.8	31.5–44.0	40.5–58.0	1/4
16.5–21.5	19.8–25.8	31.5–40.5	37.8–48.5	44.0–58.0	58.0–77.0	1/3
21.5–31.5	25.8–37.8	40.5–58.0	48.5–69.5	58.0–82.5	.....	1/2
31.5–44.0	37.8–53.0	58.0–82.5	69.5–99.0	.....	.....	3/4
44.0–58.0	53.0–69.5	.....	.....	.....	.....	1

Note: Breakdown torque range includes the higher figure down to, but not including, the lower figure.

Ratings for Fractional Horsepower Motors

Small horsepower single phase induction motors are rated primarily on the basis of breakdown torque. NEMA Standards for fractional horsepower motors are based on specific definitions of motor rating in terms of horsepower, speed, breakdown torque, service factor, and starting current.

A complete range of breakdown torque, divided into bands for the different standard horsepower ratings, is included in the Standards. Table I gives these ranges of breakdown torques for defining horsepower at different synchronous rpm speeds. Motor selection should be based on the breakdown torque required to meet the maximum peak load of the driven machine.

SERVICE FACTOR. Fractional horsepower motors having a 40 deg C rise continuous rating, referred to as General Purpose Motors, have a lower temperature rise rating than certain motors designed for specific duty and load conditions that are rated 50 deg C rise (open) or 55 deg C rise (totally inclosed).

The service factors given in Table II define the loading that will give a total temperature rise of 50 deg C on a general purpose fractional horsepower motor having 40 deg C rise at the nameplate rating. A 40 deg C motor having a service factor of 1.4 means that the motor can be operated at 140 percent of its rated load continuously.

STARTING CURRENT LIMITS. The Standard provides that the locked-rotor current of 60 cycle, single phase, 900 to 3600 rpm, motors of any type, except those split-phase motors used on washing machines and ironing machines, shall not exceed the values given in Table III. In Table IV is a list of the code letters used to designate starting current in terms of locked kva per horsepower.

Table II—Service Factor for General Purpose 40 Deg C Induction Motors

HP	Service Factor
1/20	1.4
1/12	1.4
1/8	1.4
1/6	1.35
1/4	1.35
1/3	1.35
1/2	1.25
3/4	1.25
1 at 3600 rpm only	1.25

Table III—Maximum Locked Rotor Current For 60 Cycle Single Phase Motors

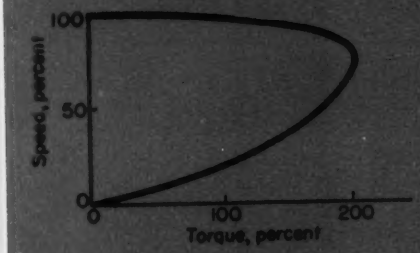
Rated Hp	Locked Rotor Current in Amperes	
	115 Volts	230 Volts
1/6 and smaller	20	10
1/4	23	11.5
1/3	31	15.5
1/2	45	22.5
3/4	61	30.5

Table IV—Code Letter Designation Of Locked KVA per Horsepower

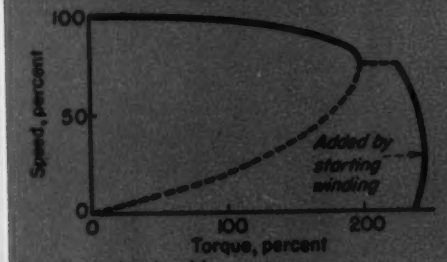
Code Letter	KVA per Horsepower*
A	0– 3.15
B	3.15– 3.55
C	3.55– 4.0
D	4.0 – 4.5
E	4.5 – 5.0
F	5.0 – 5.6
G	5.6 – 6.3
H	6.3 – 7.1
J	7.1 – 8.0
K	8.0 – 9.0
L	9.0 –10.0
M	10.0 –11.2
N	11.2 –12.5
P	12.5 –14.0
R	14.0 –16.0

* Locked KVA per horsepower includes the lower figure, up to but not including the higher figure.

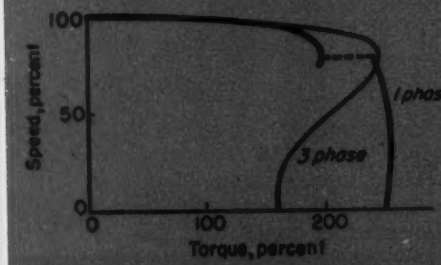
Integral Horsepower Single-Phase Motors



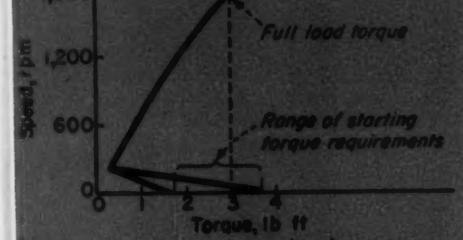
Inherent characteristic of single-phase induction motors is zero starting torque.



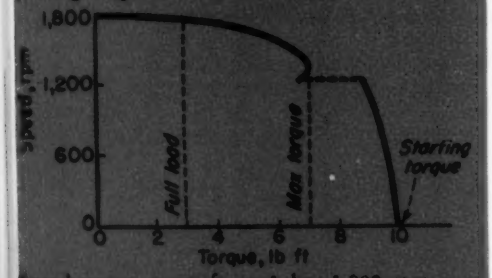
Auxiliary circuit adds starting component to single-phase induction motor curve.



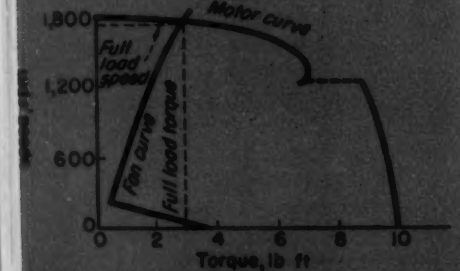
Curves of self-starting single-phase and three-phase induction motors are similar.



Speed-torque characteristic of a fan over a starting range and at full running speed.



Speed-torque curve for a 1 hp, 1,800 rpm motor at starting and at different speeds.



Speed-torque requirements of fan superimposed on speed-torque curve of motor.

Single phase induction motors have an auxiliary component circuit within the motor that starts the motor and brings it and the load up to operating speed. A speed sensitive device, voltage sensitive device, or relay disconnects the auxiliary circuit when the motor is near operating speed.

The torque produced at starting is the sum of the torque produced by the auxiliary component and the basic torque of the motor. A curve of the combined torques approximates the speed-torque curve of the polyphase induction motor. But since an extra circuit is needed inside the motor to get it started, and more material is required in a single phase motor than in a polyphase motor to obtain comparable characteristics, it is preferable to use a polyphase induction motor when polyphase power is obtainable.

CAPACITOR MOTORS have few moving parts, therefore, motor maintenance is low and mechanical wear is not severe. These motors are economical in sizes up to 5 hp at 3600 rpm.

Normal - starting - torque capacitor motors have sufficient torque to start and accelerate loads such as centrifugal pumps, fans, blowers, and woodworking machines.

High-starting-torque capacitor motors are used to drive machines that require high-starting-torque such as reciprocating compressors, reciprocating pumps, and loaded conveyor belts. If the high-starting-torque motor is used where it is not really needed, excess torque may accelerate the driven machine too rapidly and thus cause unnecessary noise and excessive wear in applications such as pumps and fans.

REPULSION-START INDUCTION-RUN MOTORS. When this motor accelerates to about $\frac{2}{3}$ of full load speed, a mechanism lifts the brushes off of the commutator and short circuits the commutator bars. The motor then runs as an induction motor.

These motors have higher starting torques than capacitor motors, but same or lower pull-up torques. Starting and running characteristics of repulsion start motors are adequate to drive most loads. They are suitable for pumps and machine tools. Electrical and mechanical noise, and extra maintenance, because of brushes and commutators are disadvantages. When the absence of noise, and accessibility of the motor are important, a capacitor motor is usually preferable.

Repulsion-start induction-run motors are applied up to 10 hp at 1,800 rpm, but from 5 to 10 hp are the motor sizes that are most generally used.

REPULSION-INDUCTION MOTOR operates as a combination repulsion and induction motor. At starting the repulsion winding supplies most of the torque, but as the speed increases the squirrel-cage induction winding becomes predominant. A mechanism is not needed to disconnect the starting circuit before operating speed is reached as is required in induction-run motors. The speed-torque curve is smooth without a break at a transfer speed. This is a distinct advantage in applications where a critical point in the acceleration of load occurs at the transfer speed.

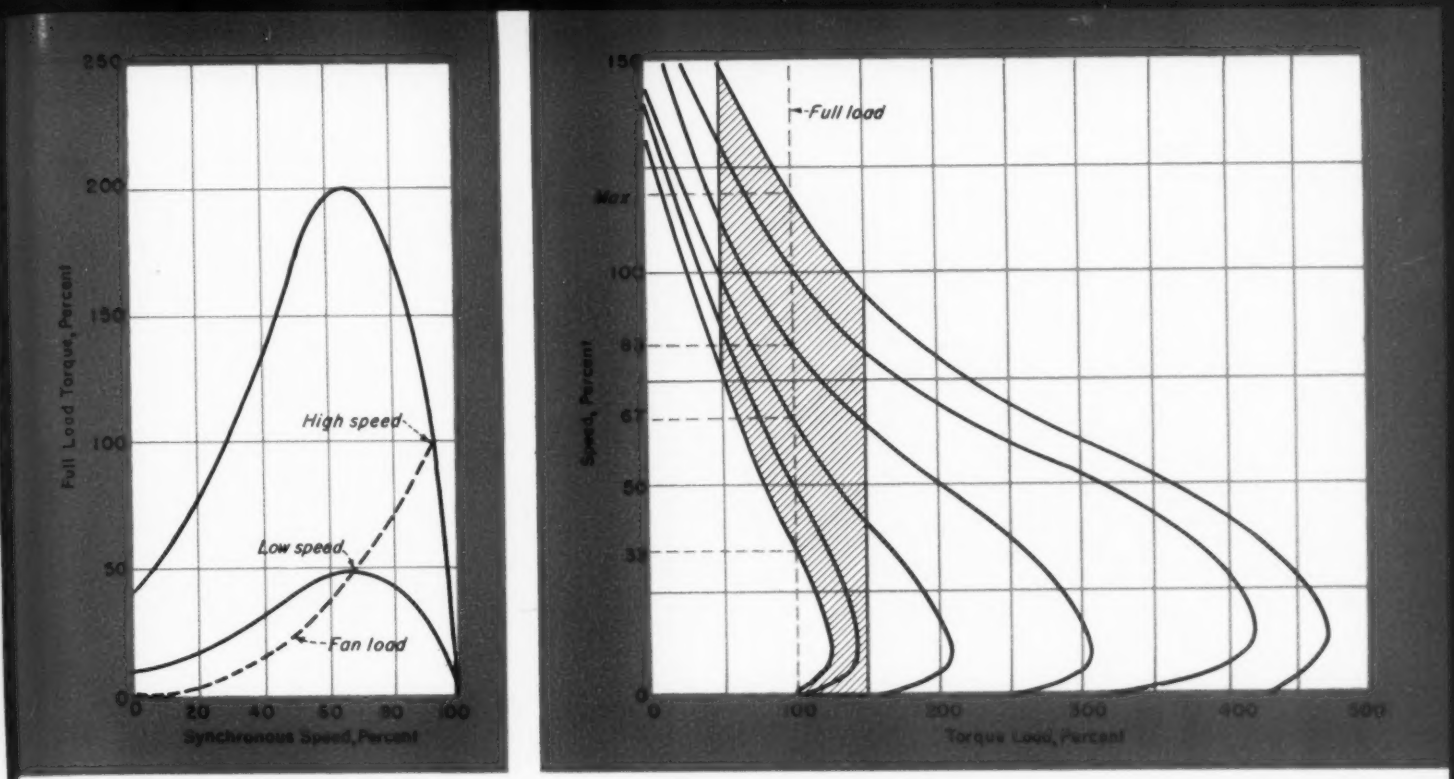
Ability of the repulsion-induction motor to withstand long accelerating periods, plus the unbroken speed-torque characteristic, make the repulsion-induction motor suitable for certain high inertia loads, such as those encountered in laundry extractors.

Since the brushes and commutator are in the circuit all the time, brush noise is continuous. This motor sometimes produces radio interference, which may be reduced by use of a filter or a capacitor.

Two types of motors for special applications are the permanent split-phase capacitor motor, and the repulsion motor.

PERMANENT SPLIT-PHASE CAPACITOR MOTOR is often used in integral horsepower sizes to drive slow-speed direct-connected fans. This motor operates like a two-phase induction motor with low starting torque, has a smooth unbroken speed-torque curve. No brushes, commutator, or centrifugal mechanism and switch are required.

The permanent split-phase motor has better than average values of maximum torque and is a superior single-



OPERATING CHARACTERISTICS (left) of permanent split-phase capacitor motor for two-speed operation. Changing position of brushes (right) on a repulsion motor gives a family of good speed-torque operating characteristics.

phase motor for momentary overloads. In spite of the many advantages of the permanent split-phase motor, its relatively low starting torque of about 30 percent greatly limits its use. Without a clutch arrangement, this motor will reliably start and accelerate only a relatively small friction load.

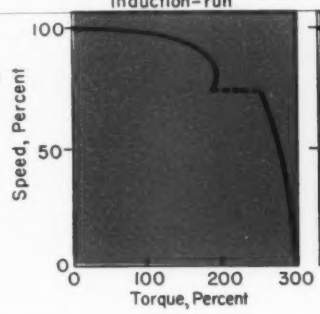
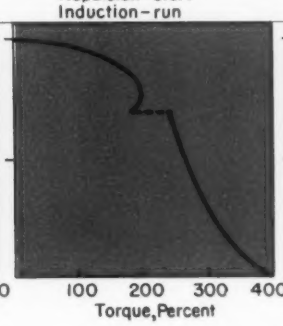
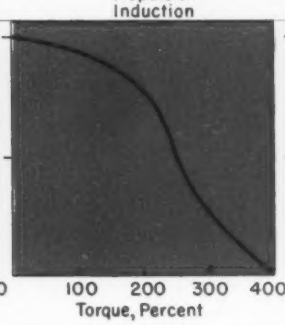
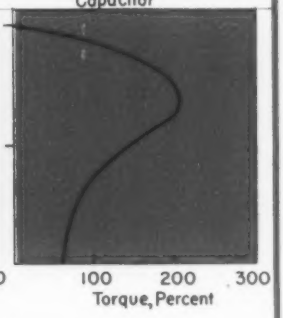
For direct connected fans, definite advantages include extreme quietness and two speed or adjustable speed operation merely by changing the voltage applied to the motor.

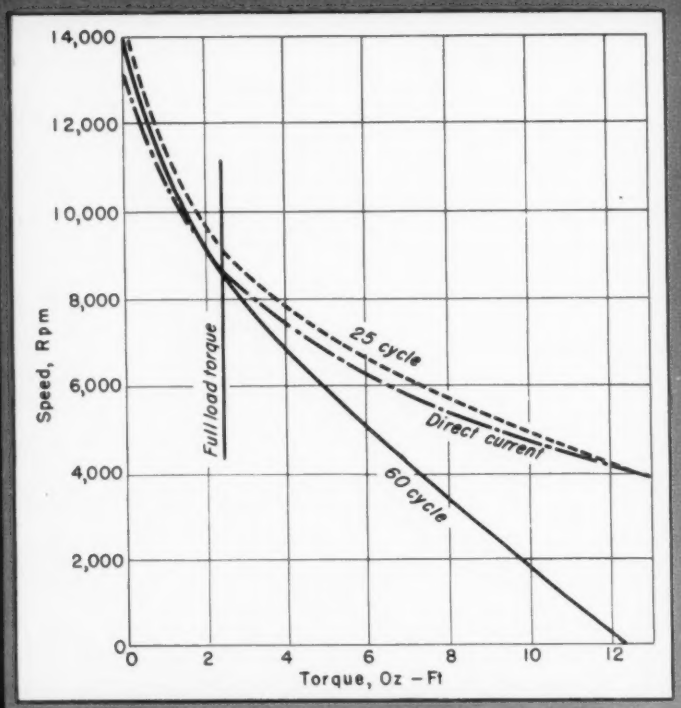
High maximum torque of this motor makes it potentially useful where heavy momentary overloads are frequent. For example, with floor sanders and grinders when the operator bears down hard on the work only high maximum torque will keep the motor from stalling.

REPULSION MOTOR. Uses the repulsion principle for both starting and running. It has the speed-torque characteristics of a series motor, with speed increasing as torque decreases. Unlike the induction motor, it has no limiting synchronous speed. Therefore its no-load speed is high, and over parts of the operating curve slight changes in torque result in large changes in speed.

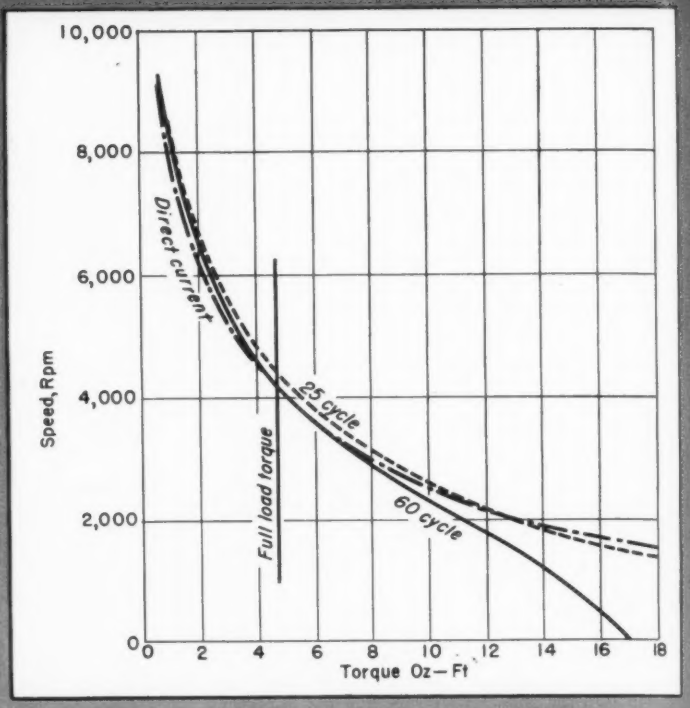
By changing the position of the brushes on the commutator a family of speed-torque curves are obtained, one for each setting of the brushes. When applied to a constant-torque load this characteristic gives wide speed control. Machines that require constant tension over a range of speeds often use the repulsion motor.

COMPARISON OF FOUR TYPES OF INTEGRAL HP SINGLE-PHASE MOTORS

Motor	Capacitor-start Induction-run	Repulsion-start Induction-run	Repulsion Induction	Permanent-split Capacitor
Typical speed-torque curve				
Standard ratings (Dripproof and totally inclosed)	1 to 3 hp, 115/230 volt, 1,800 r p m	1 to 10 hp, 115/230 volt, 1,800 r p m	5 to 10hp, 230 volt, 1,800 r p m	1 to 3 hp, 115/230 volt, 1,800 r p m
Typical applications	Compressors, pumps, fans, machine tools	Compressors, pumps, fans, machine tools	Compressors, pumps, fans, machine tools	Fans, saws
Principal advantages	Quiet operating No radio interference Low maintenance	High starting-torque Low starting-current	Lowest starting-current High starting torque Unbroken acceleration	Minimum mechanical parts Low maintenance High maximum torque
Disadvantages (Depends of quality of design)	Higher starting current	Low pull-up torque Starting noise High maintenance	Radio interference High maintenance	Low starting torque



Speed-torque curves of a 1/4 hp, 8,000 rpm, salient-pole non-compensated universal motor when operated on dc, 25 cycle and 60 cycle alternating current.



Speed-torque curves of a 1/4 hp, 4,000 rpm, compensated universal motor when operated on dc, 25 cycle, and 60 cycle, power showing improved operation.

Characteristics of Universal Motors

Speeds of universal motors are $1\frac{1}{2}$ to 10 times higher than those of induction motors. Without governor control, no-load speed may be as high as 20,000 rpm. Because power is proportional to the product of torque and speed, a universal motor occupies less space and weighs less than an induction motor of equivalent horsepower.

Universal motors will run on direct current and on alternating current of any frequency up to 60 cycles per sec. They can be wound for voltages from 32 to 250.

Characteristics of operation on alternating current are different than those obtained with direct current. For any given load, the motor runs slower on 60 cycles than it does on 25 cycles or direct current.

Speed varies with the load and voltage. Current, wattage and torque approximately vary inversely as the speed. Being a series motor, the speed theoretically will rise to infinity at no load, but windage and friction usually limit the speed to a safe value.

Efficiencies of this type motor are low when compared to those of larger motors of other types, averaging approximately 50 percent at full load.

Maximum efficiencies occur at speeds of 6,500 to 8,500 rpm. Starting torque averages 3 to 4 times the torque developed at maximum efficiency.

Constant speed can be approximately obtained from universal motors having built-in centrifugally operated governors. Governors are built for a single speed of operation or for an adjustable range of speeds. With some adjustable types, the speed can be changed with the motor either stopped or running.

For low-speed and constant-speed drives, speed reducers and governor controls can be obtained with universal motors. Worm gear or spur gear speed reducers give output shaft speeds ranging from 5 to 4,000 rpm. Such motors provide a compact, grease-tight unit for coupling to a drive shaft. Speed reducer motors are not generally recommended for continuous duty at their rated output.

A standard universal motor geared to a 5 to 1 ratio has about the same full-load speed as a 6-pole induction motor, with a $4\frac{1}{2}$-to 1 ratio the speed corresponds roughly to that of a 4-pole induction motor, and with a $2\frac{5}{8}$ to 1 ratio to a 2-pole induction motor.

Necessity for servicing brushes and commutator tends to limit the application of this motor to intermittent duty. Where operating conditions are favorable and loads light, continuous duty applications have been successful. Induction motors have a safety factor with regard to horsepower-temperature requirements that universal motors do not have.

Field of application of universal motors is limited primarily by their maximum power rating, which is about $\frac{3}{4}$ to 1 horsepower.

The characteristic of speeding up on light loads and decreasing speed as load increases, makes the universal motor desirable for applications such as portable drills, saws, routers, vacuum cleaners, sewing machines, business machines and food mixers.

Universal motors are widely used on home motion picture projectors. Some projectors have two speed governors; one speed for sound pictures and the other for silent pictures. A simple switch selects the proper governor to obtain the desired speed.

Characteristics and Types of Direct-Current Motors

Important advantages of direct current motors for machine drives are: Adjustable speed over a wide range, speeds are not limited to synchronous speeds, and variation of speed-torque characteristics.

SHUNT-WOUND MOTORS operate at approximately constant speed regardless of variations in load when connected to a constant supply voltage and with fixed field excitation. Maximum decrease in speed as load varies from no-load to full-load is about 10 to 12 percent.

Constant speed motors are usually suited for a speed range of less 3 to 1 by field control, but mechanical and electrical characteristics govern maximum safe speeds. With constant voltage on the armature, as the field is weakened the speed increases and the motor develops constant horsepower.

For speed ranges of 3 to 1 or greater adjustable voltage operation is recommended. The adjustable voltage drive usually includes a motor-generator set to supply an adjustable voltage supply to the armature of the drive motor. With this method of operation a speed range of 10 to 1 is readily obtained.

Regenerative braking can also be obtained with the variable voltage drive. This system permits deceleration of the driven load by causing the motor to operate as a generator to drive the motor-generator set, thereby returning power to the alternating current power supply.

With an adjustable voltage supply to the armature, at speeds below basic the motor is suitable for a constant-torque drive. Minimum speeds are limited by temperature rise, because the motor carries full load current at the lower speeds and at low speeds the ventilation is reduced.

Adjustable voltage drives are used for paper mills, rubber mill machinery, winders, machine tool drives and hoists.

Adjustable-speed shunt wound motors with field control are designed for operation over speed ranges of 3 to 1 or greater. Standard adjustable speed motors are rated in three ways: Tapered horsepower, continuous, 40 deg C rise; Constant horsepower, continuous, 40 deg C rise; and Constant horsepower, 1 hr, 50 deg C rise, of the next larger horsepower rating than for continuous duty.

One hour 50 deg C open motors develop constant horsepower over the entire speed range. Semi-inclosing covers can be added without changing the rating.

The 40 deg C continuous rated open motors develop the rated horsepower from 150 percent of minimum speed to the maximum speed. From minimum speed to 150 percent of minimum speed, the rated horsepower will be developed continuously without exceeding safe temperature limits.

Tapered horsepower motors develop the maximum rated horsepower at 3 times the minimum speed. The horsepower decreasing in direct proportion to the decrease in speed down to the horsepower rating at 150 percent of the minimum speed.

SERIES WOUND MOTORS are inherently varying speed motors with changes in load. On light or no loads, the speed may become dangerously high. These motors should be employed only where the load is never entirely removed from the motor. They should never be connected to the driven machine by belt.

Series motors are used on loads that require very high starting torques or severe accelerating duty, or where the high-speed characteristic may be advantageous, such as in hoists.

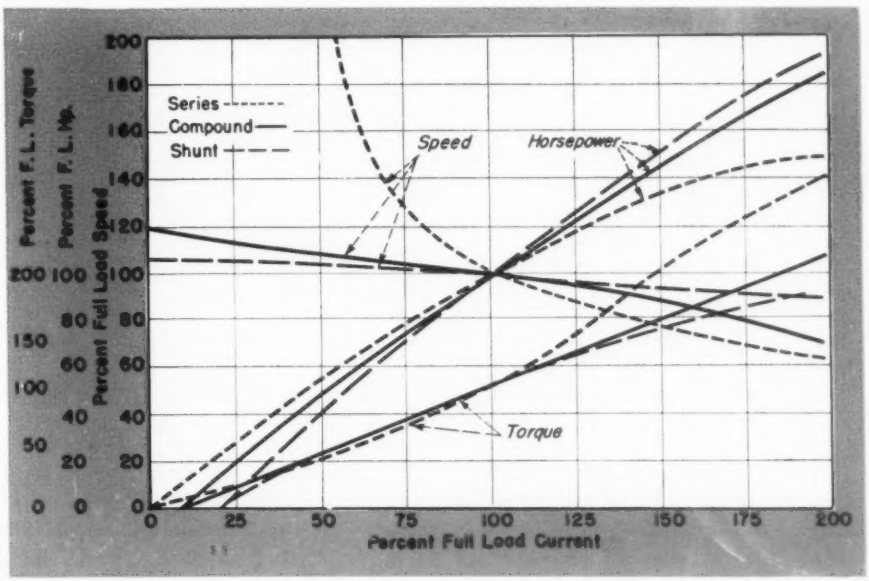
COMPOUND-WOUND MOTORS are used to drive machines that require high starting torque or in which the loads have a pulsating torque. Changes in load usually produce wide speed regulation. This motor is not suited for adjustable speed by field control.

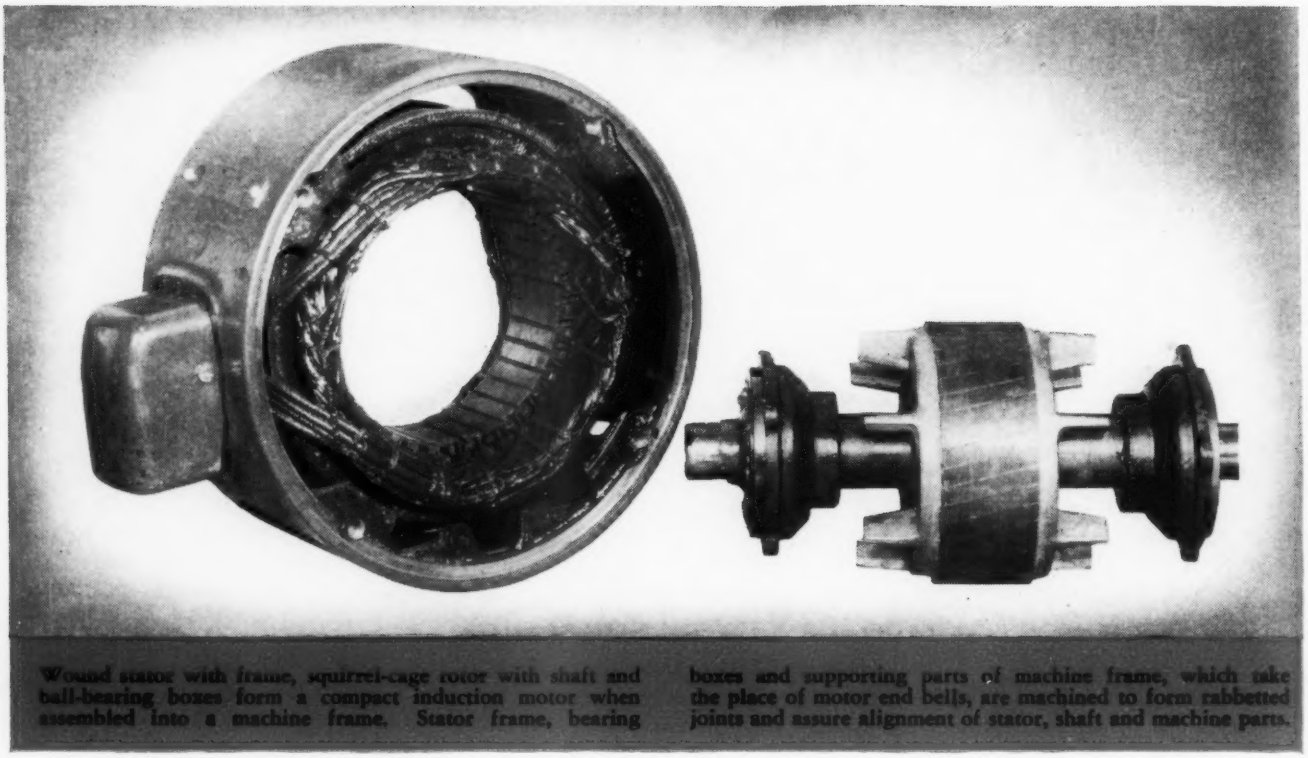
The relative strength of the shunt and series fields of the motor determines to what extent the motor approaches the shunt or series characteristics.

From no-load to full-load, the drop in speed of compound wound motors is approximately 25 percent.

Compound-wound motors are used where reasonably constant speed is required; and for loads where high starting torque is needed to accelerate the driven machine.

Compound-wound motors should be used on machines having flywheel or high inertia loads, wherein the drooping speed characteristics of the motor causes the flywheel to give up its energy as the shock load comes on; thereby cutting down the power peaks taken from the line, and resulting in less heating of the motor.





Wound stator with frame, squirrel-cage rotor with shaft and ball-bearing boxes form a compact induction motor when assembled into a machine frame. Stator frame, bearing boxes and supporting parts of machine frame, which take the place of motor end bells, are machined to form rabbetted joints and assure alignment of stator, shaft and machine parts.

Built-In Shell Type Motors

Close center dimensions in multiple shaft or spindle machines, can be obtained by using built-in polyphase squirrel cage induction motors, which consist of a shaftless rotor and a shell that contains stator punchings and windings.

Compactness can be achieved by keeping over-all diameter of motor small and using longer parts than those built into standard motors.

Operation at speeds greater than 3600 rpm is possible with rotors of small diameter.

Multispeeds can be obtained by using a multifrequency power supply.

Designer of the driven machine must make arrangements for bearings, lubrication, ventilation, and balancing at critical speeds.

Alignment of rotor shaft and stator with machine into which shell type motor is built is usually provided by joints of rabbetted construction.

Applications for shell-type, high-speed motors are:

Boring machines, multiple-spindle drilling machines, grinders, saws, wood-working machinery, rayon and thread spinning machines.

Partial Motors

Simple and compact drive arrangements are often possible by using parts of motors to make an integral drive and machine assembly.

Driving unit, consisting of a motor minus end shield and bearing, can be bolted directly to the end or side of driven machine, and machine shaft coupled to motor shaft.

Short over-all length of driving unit and machine are possible by using motor parts as components of the driven machine.

Compactness in design can be achieved by using motors minus end shields, bearings and shafts, with rotor mounted directly on shaft or spindle of driven machine.

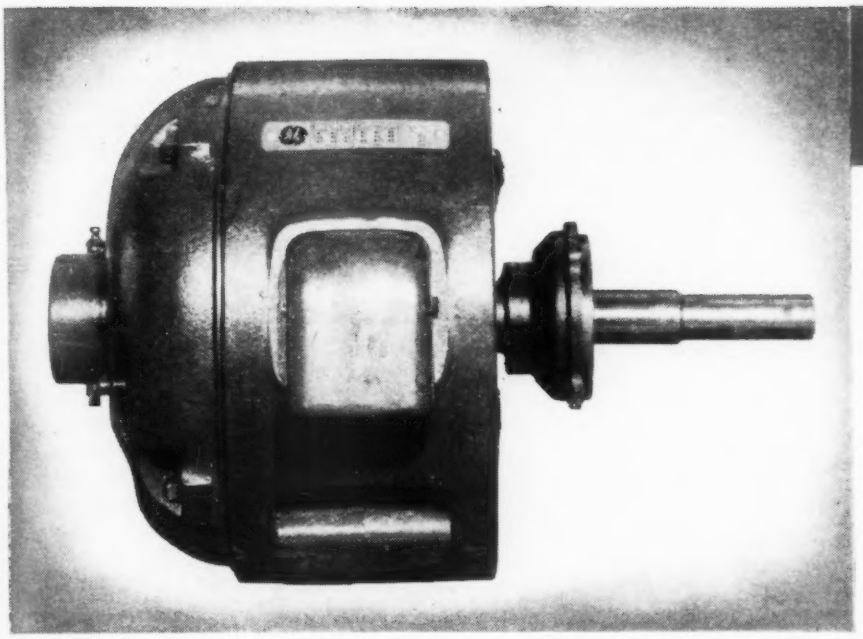
Means for proper ventilation, mounting, alignment and support of bearings when using partial motors, must be provided in the design of the machine.

APPLICATIONS: Machine tool head stocks, small hoists, gear reducers, grinders, compressors.

Built-In Motor Parts

CASE HISTORY. A cam milling and grinding machine was originally powered by four motors, including a 2 hp. motor on the cutter spindle, a 1 hp on the work head, a $\frac{1}{2}$ hp for the rapid traverse of the work head and a $\frac{1}{6}$ hp for the chip blower. The original machine had the main drive motor mounted on a bracket on the back of a column with a flat belt running to a jack shaft supported in a pivoted bracket. A chain drive transmitted this power from the jack shaft to the cutter head. This method of driving the cutter head was used for many years on this machine, but has certain disadvantages including high maintenance and first cost as well as an unsatisfactory appearance.

The manufacturer of this machine decided to redesign the main drive, which consists of the cutter head, to accommodate a rolled shell type, shaftless, built-in motor which would drive the cutter head spindle direct through a high speed gear shaft. This redesign did away with the mounting for the standard motor on the back of the column, all of the pulleys, belts, jack shaft and the chain drive. Power was transmitted to the built-in motor on the cutter head through a flexible cable. This new arrangement with the built-



Round frame, ball-bearing squirrel-cage induction motor without the back end shield can be mounted in an overhanging arrangement on the frame of driven machine.

in motor gave greater flexibility in design of the machine, reduced the maintenance and most important of all, resulted in a considerable saving in cost of the machine.

CASE HISTORY. A high speed turret lathe was originally driven by a standard motor in the base of the machine. Motor was belted to a step pulley spindle.

Machine was completely redesigned around a rolled shell type, shaftless, 4-speed motor. Rotor was mounted directly on the main spindle.

Drum controller for speed setting and a lever operated start-stop and reverse were built into the machine head. Motor was designed for frequent reversing duty and gives instant response in operation to a degree that was impossible in the original belted spindle.

A high degree of balance was obtained in this machine by using only one rotating unit.

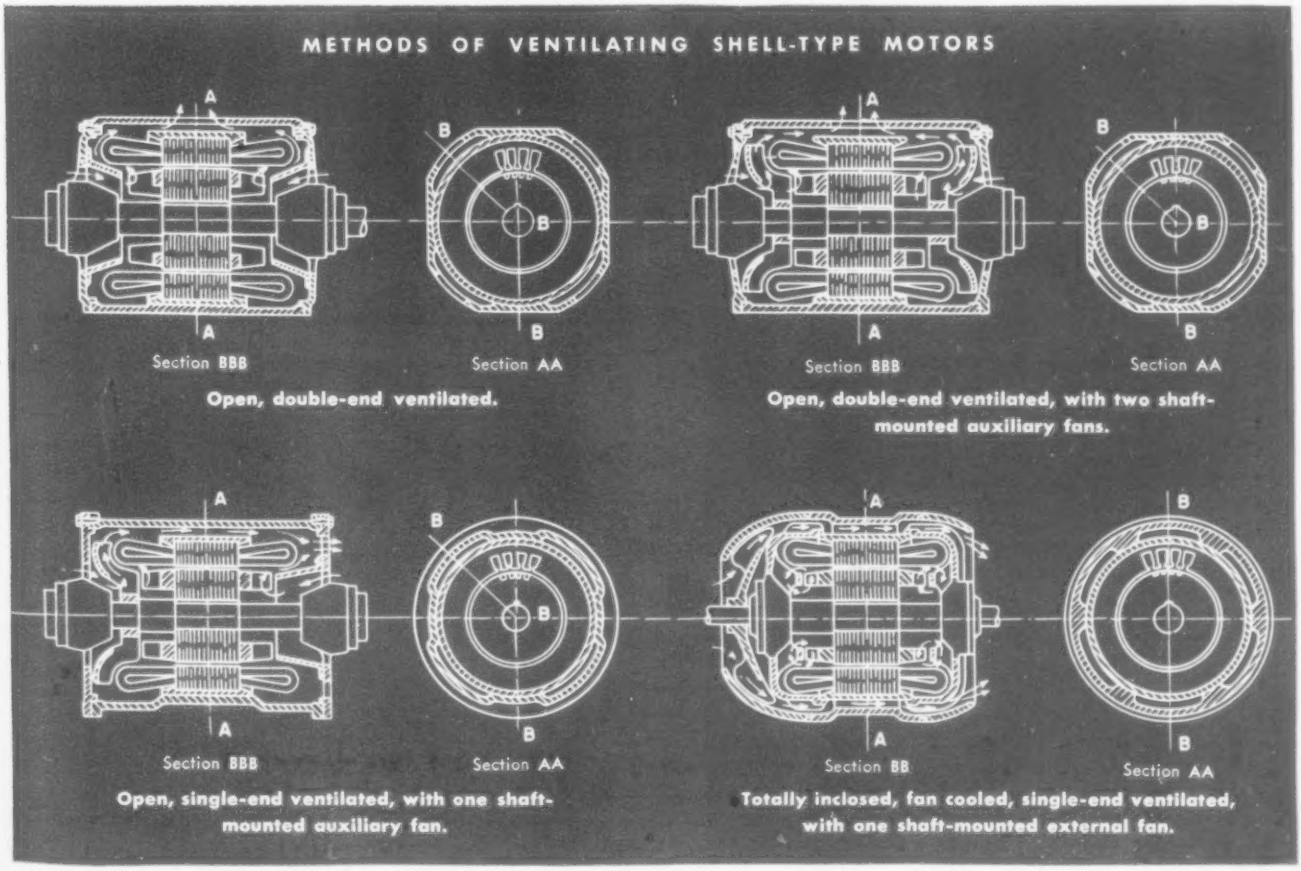
Flange-Type Motors

No base plate needed for support of driving unit when footless round frame motors with flange-type end shields are bolted directly to the driven machine.

By mounting a flange-type motor with a suitable speed directly on the driven machine, it is often possible to save not only the cost of a bedplate extension for the motor but the cost of an intermediate transmission drive as well.

No support needed for many types of small or self contained machines that can be bolted to or overhung on flange-type end shield motors that have feet.

Accurate alignment of motor and


METHODS OF VENTILATING SHELL-TYPE MOTORS
A
B
Section BBB
Section AA
Open, double-end ventilated.
Open, double-end ventilated, with two shaft-mounted auxiliary fans.
Open, single-end ventilated, with one shaft-mounted auxiliary fan.
Section BB
Totally inclosed, fan cooled, single-end ventilated, with one shaft-mounted external fan.

driven machine is assured by rabbett in mounting surface of end shield.

Tight-fitting coverplate for machine is provided by solid face of end shield.

Motor shaft extension can be made to suit specifications.

Compact direct drives are possible, without the necessity of the designer providing arrangements for ventilation, lubrication, bearing supports and alignment of connected parts.

No cooling air comes from the driven machine. Ventilating air inlet to motor is on the motor side of end shield face.

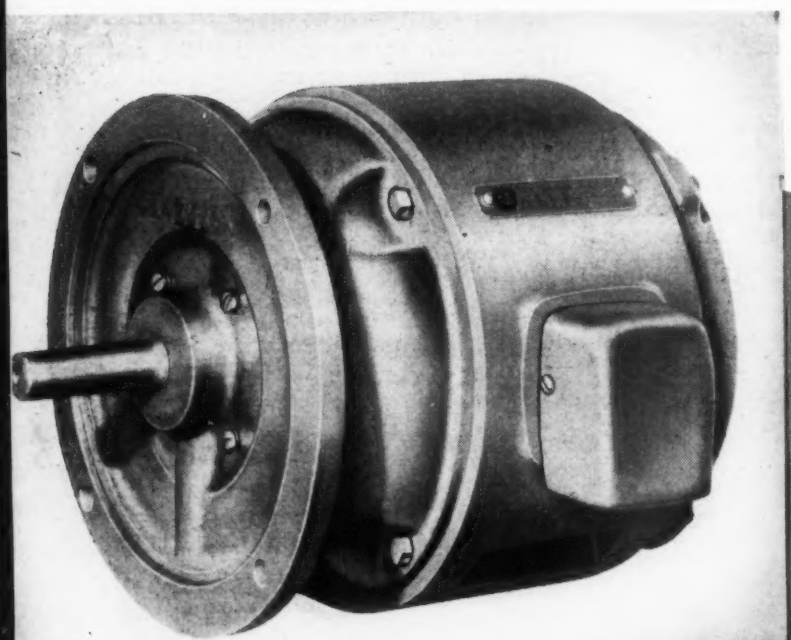
Different installation requirements can be met with little or no change in the design of the machine, since motors of different power and service ratings or types of inclosures can be furnished with the same end shield mounting dimensions.

With special shaft extensions, two separate rotating parts may be supported and driven by a foot mounted, flange-type end shield motor. An oil burner assembly with fan and pump mounted on the motor shaft is an example of such an application.

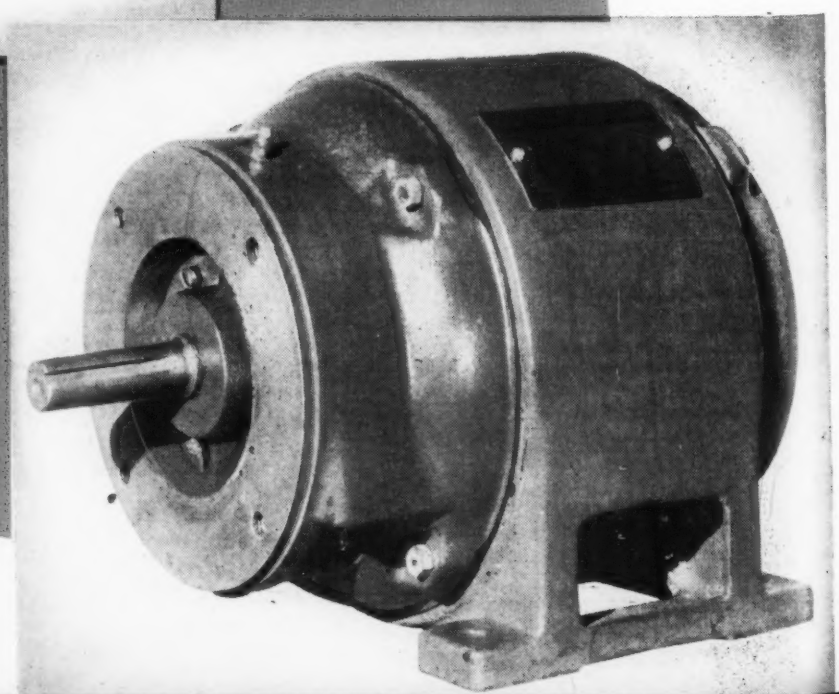
Smaller high speed rotating parts, with consequent reduction in weight, can usually be obtained with built-in motor parts.

Cost of the complete machine can often be reduced, space conserved, and appearance improved by building standard motor parts into the machine. These parts make possible a wide variety of integral motor mountings. Parts are available that will meet the requirements of many conditions of service, such as totally inclosed fan cooled, totally inclosed non-ventilated, drip proof, semi-inclosed, splash proof, and pipe-forced ventilated.

PRODUCT ENGINEERING thanks the following companies for their helpful cooperation in contributing material and data for the preparation of this special editorial section: Alliance Manufacturing Co., Allis-Chalmers Mfg. Co., Bijur Lubricating Corp., Bodine Electric Co., Emerson Electric Mfg. Co., General Electric Co., John Oster Manufacturing Co., Leland Electric Co., Louis Allis Co., Mercury Clutch Corp., Reliance Electric & Engineering Co., Twin Disc Clutch Co., Westinghouse Electric Corp.



Ball-bearing, squirrel-cage induction motor with flange type rabbetted end shield for direct bolting to the frame of the driven machine.



Foot mounted, squirrel-cage induction motors with face-type end shields can be used for driving and supporting many types of small machines.

Electroforming Parts with High Strength

From "The Electroforming Process" by William Orbaugh, *The Rose Technic*, Volume LVIII, No. 3, October, 1947.

ELECTROFORMED PARTS are made by electrodeposition of metals over the surface of a mold or matrix to a thickness of 0.010 in. or more. The chief advantages to be gained from electroforming are: (1) High dimensional accuracy, (2) fine internal surface finishes. In addition, the physical characteristics of electrodeposited metals are excellent.

Electroforming was originated over a hundred years ago by Jacobi for making electro-type printing plates. Later the process was used to make seamless copper tubing and copper sheet. More recently it has been used to produce floats, phonograph record masters, gaskets, fine mesh screens, searchlight reflectors, and leather embossing dies. Wartime developments accelerated electroforming techniques. Radar guides, almost impossible to produce by any other method, were electroformed. These guides required complicated shapes, interior surface finish from 2 to 5 rms and overall dimensions accurate to within ±0.001 inch. Other applications were for forming three dimensional cams, computing-pin cams, and venturi tubes. At the present time it is also being used to plate metal molds for plastics.

STRENGTH OF ELECTRODEPOSITED METALS

The strength of electroformed parts is high. Metals usually used for electroforming are copper, nickel and copper. These materials, used singly or in combination, offer a wide range of properties to meet the designers' requirements for corrosion resistance, strength and wear. Internal stresses for electroformed parts are lower than those present in many welded, forged, or cast sections. In addition physical properties equal, and sometimes exceed, those for the same metals in the cast, rolled or forged condition.

Electroformed copper has tensile strengths from 20,000 to 50,000 psi, however, the Brinell from 40 to 150, tends to limit the number of applications for this material.

Electroformed iron in the as-plated condition has a tensile strength of 50,000 psi, Brinell of 230, and an elongation of 15 percent. In the fully annealed condition, the tensile strength is 50,000 psi, Brinell drops to 125 and elongation rises to 30 percent. Iron parts may be carburized and heat treated to produce any desired hardness from 16 to 62 Rockwell C.

Nickel may be deposited with tensile strengths ranging from 50,000 to 150,000 psi depending on the type of bath and plating conditions. The table below lists physical properties of nickel deposited by various baths.

By varying bath temperature and density, intermediate physical properties can be obtained.

Fig. 1 shows some of the physical properties of a nickel produced in an all-chloride bath and annealed from various temperatures.

Electroformed parts may be laminated by depositing two different metals to obtain certain desirable characteristics. It is interesting to note also that all of the common metals used in electroforming may be readily machined.

ECONOMICS OF ELECTROFORMING

It is difficult to compare the economic advantages over other methods of fabrication unless the cost of producing individual products are compared. Some items, previously discussed, cannot be economically or satisfactorily produced if made by other processes. Equipment cost for

Table I—Physical Characteristics of Electrodeposited Nickel

Plating process	Soft bath	Medium hard nickel bath	Double-salt hard bath
Ultimate tensile strength, psi	50,000	100,000 to 120,000	120,000 to 150,000
Hardness, Brinell	150	260 to 280	.....
Rockwell hardness, C	.....	.....	28 to 45
Elongation, percent	30	12 to 20	.....

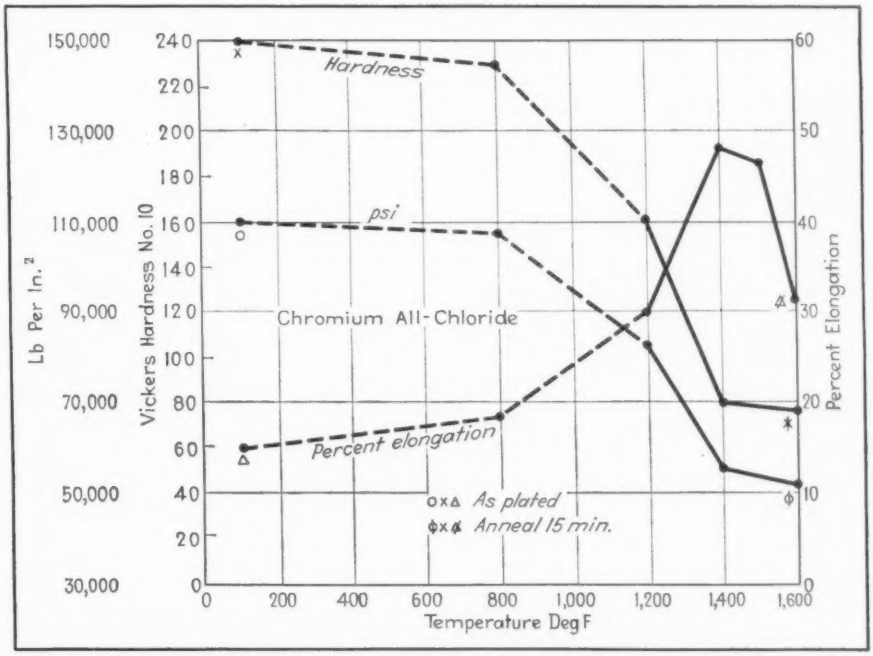


Fig. 1—Effect of annealing temperature on deposit from all-chloride nickel bath of pH = 2.0 at 55 asf. Temperatures over 800 deg F have great effect on physical properties.

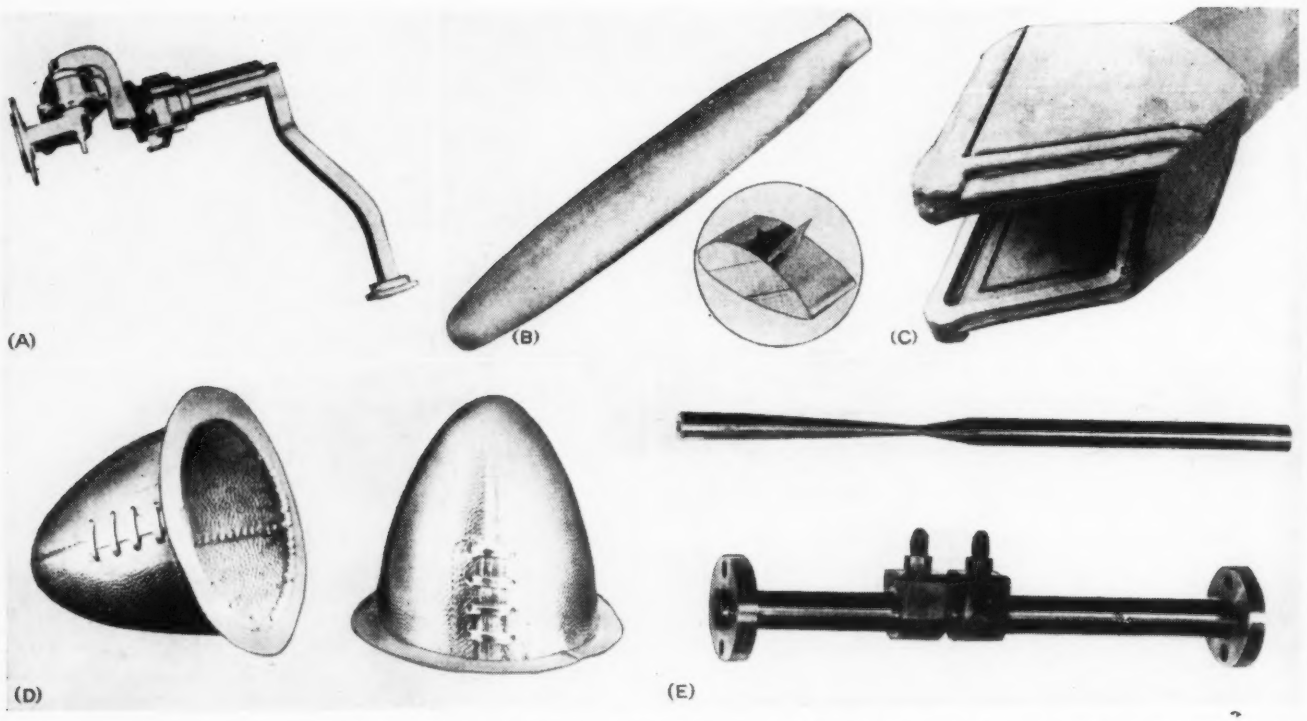


Fig. 2—Various parts made by electroforming. (A) Rotating-Joint radar wave-guide assembly made in three sections. (B) Nickel plating wood propeller blade makes it stronger and more weather resistant. (C) Jet propulsion motor body requiring precise dimensions. (d) Electroformed male mold for precision casting football mold. (a) Aluminum core (top) and formed venturi meter.

electroforming is considerably lower than that for machine tools used to produce the same dollar volume of work.

Design of Electroformed Parts

Electroformed parts should be designed so that the mold or matrix, upon which the deposit is built up can be readily removed. Slight tapers will allow easy removal of the mold. Rounded corners are preferable to sharp curves and sharp, deep recesses should be avoided. Because the surface of the electroformed part duplicates the surface of the mold, any surface texture may be obtained depending on that put on the mold surface.

Additional applications for the electroforming process are resizing of worn or mis-machined parts and heavy metal coatings on non-conductors. This process is especially suitable in building up shafts because the build-up can be made very hard without affecting the softer shaft material. Non-conductors that have been plated include plastic impregnated antenna masts and wood aircraft propellers. The metal plating has greatly improved the strength and weathering properties of these parts.

Creep Rate of Various Industrial Leads

Foreign abstract condensed from "Creep Rate of Various Industrial Leads", by J. Neill Greenwood and J. H. Cole, *Metallurgia*, September, 1947.

Investigations show that traces of some elements are effective in changing the creep rate of lead, some increasing and others decreasing the rate. Industrial leads from various sources and refined by different processes were examined to see whether the composition differences gave results in agreement with those obtained from synthetic laboratory products.

The influence of a given element was found to be not necessarily progressive. In some cases (copper and bismuth) the first additions caused an increase in rate, while further additions reversed this tendency. The opposite effect was also found (silver), namely, that the improvement with small additions was lost with further increase in concentration.

A summary of the rate of creep of different lead alloys at 20 C, stress 350 lb per sq in., condition annealed at 100 C for 15 min, shows that the elements normally present in lead as impurities, namely, silver, antimony and bismuth do not greatly influence the creep rate when present up to 0.10 percent. Zinc, 0.05 percent, appears to reduce the creep rate of lead to about one-fourth; copper over 0.05 percent to about one twenty-fifth; and calcium, 0.05 percent, to one three-hundredth of the rate for "U" lead. The influence of calcium, therefore, is quite outstanding.

The material for test was prepared in the laboratory by rolling 6x1x1 in. ingots to strip $\frac{1}{8}$ in. thick at a temperature of 20 to 24 C. Each pass reduced the thickness by 0.028 inch.

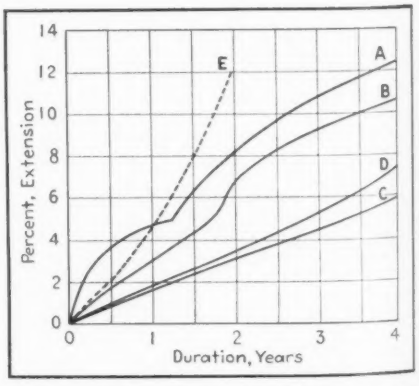


FIG. 1—Creep curves for different leads.

The strips were tested in two conditions: (1) As rolled, and (2) annealed 15 min at 100 C in an air oven. Annealing was done before the strips

TABLE I—Analyses of Leads Tested

Element	Additions, hundredths of a percent							
	1	2	3	4	5	6	7	8
Cu	0.01	0.02	0.02	0.05	0.02	0.02	1.40	0.06
Bi	0.07	0.23	0.14	0.11	0.72	4.93	0.00	1.25
Cd	0.01	0.01	0.01	0.01	0.01	0.01	nil	0.02
Sn	nil	nil	nil	nil	nil	nil	nil	nil
Sb	0.05	0.42	0.00	0.21	0.34	0.29	0.95	0.03
As	nil	nil	0.07	0.02	0.10	0.00	0.00	0.00
Fe	0.06	0.07	0.02	0.07	0.05	0.04	0.03	0.06
Zn	0.02	0.05	0.02	0.04	0.03	0.04	0.07	0.04
S	0.01	0.04	0.03	0.04	0.07	—	0.56	0.04
Ni	—	—	0.00	0.00	nil	0.00	0.14	0.01
Ag	0.00	0.03	0.03	0.06	0.04	0.02	0.40	0.28
Ca	—	—	0.05	—	—	—	—	—
Total	0.23	0.87	0.39	0.61	1.38	5.35	3.55	1.79

were cut to shape. Two stresses were imposed, 500 and 350 lb per sq in. respectively at 20 C temperature.

Typical creep curves are given in Fig. 1 for the leads listed in Table I. Curves *A* and *B* are characteristic of the leads designated 1 to 6, and curves *C* and *D* are for the leads designated as 7 and 8 respectively. Curves *A* and *B* show that there is a distinct increase in the rate of creep during the recrystallization period. Recrystallization under stress was noted with all the leads except numbers 7 and 8. The curves for numbers 7 and 8, as given in Fig. 1, reveal an important difference in behavior. For the first year and a half under stress, the two curves run almost parallel and the values of total increase in length are almost identical From this stage on the two curves diverge, the creep rate of number 8 increases continuously and that of number 7 decreases continuously. These features are characteristic of the influence of silver and copper respectively.

The authors conclude that:

1. The total impurity content of industrial lead is no guide to its behavior under prolonged stress.

2. The nature of the impurity and its specific influence must be known. Even then the influence of one impurity may be affected by the presence of others.

3. It is suggested that for the purposes of a first classification of industrial lead alloys a test of 100 days' duration under a stress of 500 lb per sq in. may be used, the time to extend 2 percent being the criterion.

4. To supplement the latter, the test piece at the end of 100 days should be examined for embrittlement by bending and by micro examination. The latter will reveal any incipient intercrystalline weakening.

Properties of Some New Insulating and Magnetic Materials

From symposium of conference papers on materials presented at 1948 Winter Meeting AIEE, Pittsburgh, Pa.

Polymers For Insulation, by J. R. Perkins, E. I. du Pont de Nemours.

Teflon (polytetrafluoroethylene) is composed entirely of fluorine and carbon. Major uses are as replacement for mica filled tape and as primary insulation for wire and cable. Teflon characteristics include: (1) Can operate continuously at 200-250 C without deterioration; (2) not attacked by chemicals or solvents; (3) extremely low electrical losses at all frequencies, including micro waves; (4) negligible water absorption; (5) and does not form carbonized path in presence of continuous arcing. Fabrication is limited to compression molding and extrusion of simple shapes. Tape is made by shaving a molded cylinder.

Polyethylene has similar physical and electrical properties to those of Teflon at approximately room temperatures. Depending upon load, its use is limited to 85-100 C. Uses include spaghetti tubing, electrical tape, wire insulation, potting compounds, and the material looks promising as dielectric for capacitors. Polyethylene is: (1) Lightest commercial plastic; (2) relatively inexpensive; (3) easily molded by injection or compression methods, and extruded.

Nylon is used: (1) For mechanical parts such as gears, racks, knobs, bushings and bearings for light loads; and (2) for electrical insulation as tapes, tubing, wire covering and cable sheathing. Nylon is extremely tough and abrasion resistant, has a very high melting point, and excellent chemical resistance. As a molding powder, it can be injection molded or extruded as sheets, rods, or tubes.

BCM, developed as a low pressure laminating resin, is particularly suited for two stage laminating work. BCM is promising as thermosetting molding composition using more than 50 percent of filler with short molding cycles. Injection molding with a cold cylinder and a hot mold on 15 sec cycle cure seems practicable. Compression is likewise more rapid. Use of asbestos filler should simulate desirable properties of cold molded compounds plus better dimensional tolerances.

Polyester Resins, by C. F. Hill and N. C. Foster, Westinghouse Research Laboratories. Solvent reactive-types of insulating varnishes differ from conventional varnishes in that the solvent reacts with the base resin and becomes a part of the final solid. Thus, because of essentially complete filling, thermal transfer, heat resistance, dielectric strength, and heat stability often are improved. The resins are used for surface coating or casting, or for impregnating. In hardness, they range from glass-like to soft. Electrically, the best are about equal to the usual high grade synthetic solvent-type resins. But because these varnishes have relatively short tank life, special techniques sometimes are necessary for best results.

Permafil by J. A. Loritsch, General Electric Co. Permafil is an impregnating and insulating material with: (1) Completely reactive ingredients and no inert solvent; (2) through-curing with no volatile by-products; and (3) through-curing without need of oxygen. Permafil makes possible coil structures that are corona free, have relatively constant power factor over wide range of temperature and voltage, resist moisture, and on d-c dissipate more than normal wattage.

Titanate Dielectrics by G. R. Shelton, National Bureau of Standards. Some ceramic bodies, particularly the

titanates, are conspicuous because they exhibit high values of the dielectric constant. Such bodies are being produced commercially as dielectrics for instruments used in radio, radar, television, and hearing aids. Current investigations on ceramic dielectrics include the determination of properties of titanates of the alkaline earth oxides. In the preparation of these dielectrics, mixtures in the proper proportions of titanium dioxide and carbonates of the alkaline earth elements were calcined and pulverized. The resulting materials were dry-pressed, at 20,000 pounds per square inch, in the form of disks $\frac{1}{8}$ in. in thickness and $\frac{1}{4}$ to $1\frac{1}{2}$ in. in diameter. The disks were heated to maturity as indicated by 0.1 percent of absorption. Results were obtained for heat treatments, absorption, shrinkage, thermal expansion, dielectric constant K at —60 degrees to 85 C, also Q-values at 25 C for frequencies of 50 to 20,000 kilocycles per second. Some measurements of K and Q were made at 3,000 megacycles per second.

For dielectrics with compositions in the seven systems of oxides, the ranges in values of K and Q, measured at 25 C and 1 megacycle per second, are:

System	*K*	*Q*
MgO-TiO	12-97	69-10,000
CaO-TiO	60-144	350-10,000
SrO-TiO	97-260	2,600-8,000
BaO-TiO	34-1,500	50-5,000
BaO-MgO-TiO .	12-1,550	9,10,000
BaO-CaO-TiO .	35-2,700	30-10,000
BaO-SrO-TiO ..	34-18,000	50-10,000

MAGNETIC OXIDES by F. G. Brockman, Philips Laboratories, Inc. A new development in soft ferromagnetic materials, called "Ferroxcubes" have high electrical resistivity as compared with the usual magnetic metals and alloys. Thus eddy current losses are extremely small since these solid solutions of ferrites have 10^7 to 10^{12} times the resistivity of iron.

Although numerous varieties are in the developmental stage, one in limited production is Ferroxcube III. Typical characteristics are:

Initial permeability, μ_0, 1,000 to 1,500

Losses, at 60 kc, tan δ/μ, 0.08 to 0.12×10^{-4}

Hysteresis coefficient, at 2 kc, α, 2 to 5×10^{-6}

The hysteresis coefficient α is that used by Legg.

In these data the quantity tan δ/μ is the phase angle between B and H divided by permeability. It is related to the quantity $R/\mu fL$ used by Legg ("Magnetic Measurements at Low Flux Densities Using the A-C Bridge," Bell System Technical Journal, 1936), by the relationship:

$$\frac{\tan \delta}{\mu} = \frac{1}{2\pi} \frac{R}{\mu f L}$$

SPECIAL MAGNETIC ALLOYS by G. W. Elmen and E. A. Gaugler, Naval Ordnance Laboratory. Magnetic cores made of materials with substantially rectangular hystersis loops increase the power output of such apparatus as mechanical rectifiers and magnetic amplifiers. The material is especially promising in the design of saturable devices operating at high flux density.

WELDING GLASS ELECTRICALLY by E. M. Guyer, Corning Glass Works. Depending upon temperature, glass can be heated electrically in three ways: (1) At low temperature where glass is an insulator, high frequency dielectric heating is effective; (2) at moderate temperatures where glass behaves as electrolyte, electrical conduction is efficient and inexpensive; and (3) at high temperatures, conducting glasses with low viscosity can be heated by high frequency induction heating.

A newly developed high frequency torch using electrically conducting needle flames makes possible fabrication of glass parts that previously were not practicable. For example, designs produced include miniature opticle cells, 20 in. television tubes, glass doughnuts for multimillion volt electron accelerators, and many miles of field fabricated Pyrex pipe line. The pin point fires preheat localized areas until the glass becomes an electrical conductor, then high frequency current is passed through the flames into the glass, and the operation is completed by electric melting.

Compared to conventional flame sealing, advantages of electrically welding glass include: (1) Deep penetration into thick sections of high melting-temperature glass; (2) rapid attainment of high temperature without destructive boiling of surface; (3) precisely placed localized heating without distorting adjacent glass; (4) better temperature control and more uniform results; and (5) faster heating.

Chill-Cast Tin Bronze

From "Tin and Its Uses, No. 18," Tin Research Institute, England. Copies can be obtained from the Battelle Memorial Institute, Columbia, Ohio.

MODIFICATIONS in the methods of manufacturing chill-cast tin bronze arising from recent researches, have led to appreciable economies in production costs as well as to considerable improvements in the qualities and mechanical properties of the product.

Bronze containing 8 to 14 percent tin can now be fabricated by hot-working methods, such as forging, stamping or extrusion, or may be cold worked by rolling or drawing.

Solid drawn tubes can be produced under ordinary industrial conditions. These tubes can be used in condensers and in all types of heat exchange apparatus.

The corrosion resistance of bronze is already well established. A series of natural color photographs shows its resistance to sea-water impingement, sedimentation and corrosion tests representative of very severe operating conditions.

Extensive trials in laboratories and on an industrial scale under the most rigorous conditions have proved that the new bronze tubes provide engineers with a material that will give excellent service under exacting circumstances.

Magnesium Alloy Progress

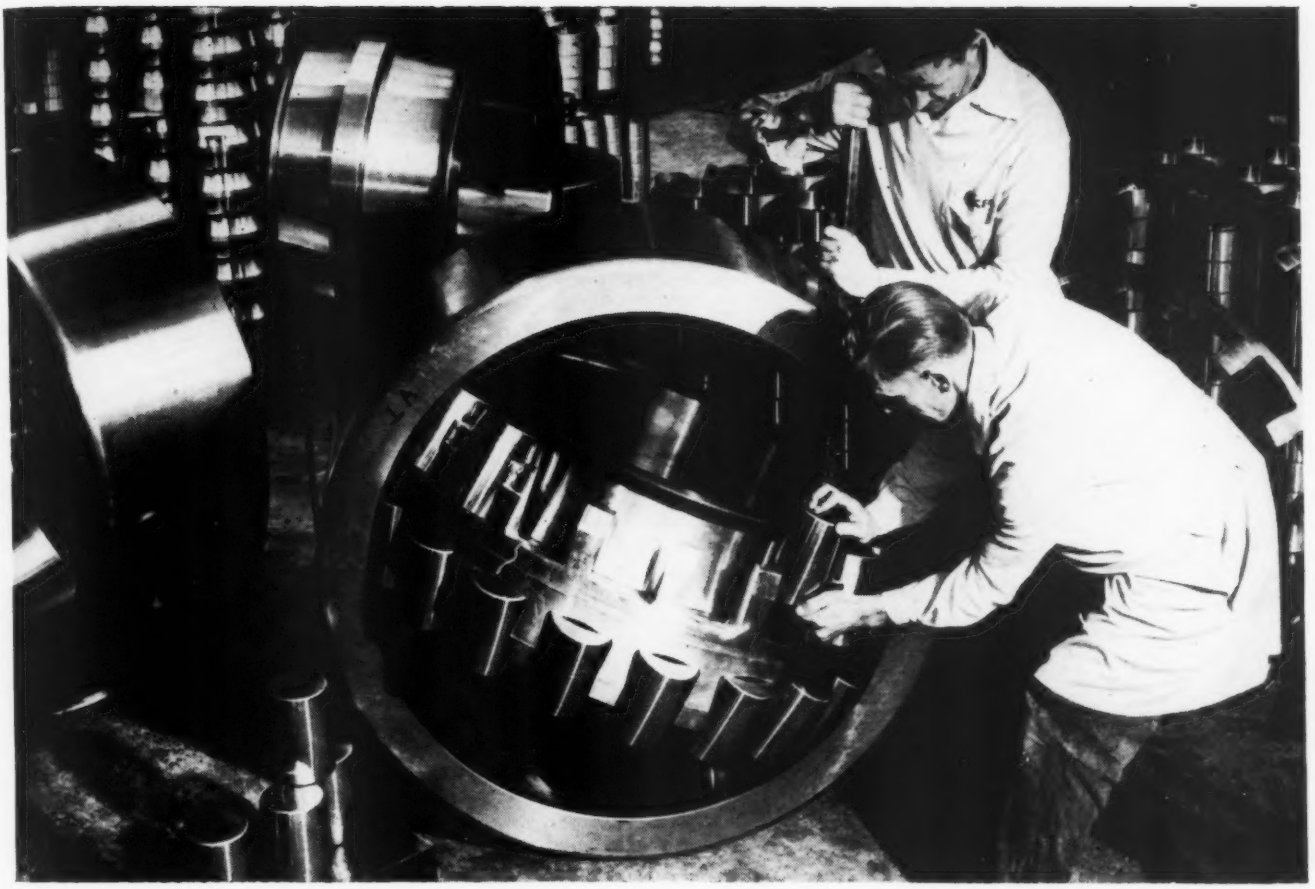
From "Magnesium and its Alloys", by R. G. Wilkinson, *Metallurgia*, January 1948, Vol. 37, No. 219, page 133.

MAGNESIUM ALLOY APPLICATION DEVELOPMENT, and restoration of applications used prior to World War II, have caused steady increase in magnesium use. The main factor in this expansion (British) has been establishment of ingot production and foundry techniques capable of making casting alloys at lower prices and in greater quantity.

Germans developed use of forgings to an advanced stage prior to and during the war. Forgings of zirconium alloys have indicated possible uses at the elevated temperatures of jet engines. Aluminum-containing alloys may be replaced by zirconium alloys in many forging applications where resistance to high temperature is necessary.

Welding of magnesium alloys has shown no spectacular advance over the past year, but steady development of techniques continues. The new noncorrosive fluoride base flux is being used increasingly in gas welding processes. This flux has a high melting point and cannot be used with alloys that contain aluminum or zinc. Weld residue need be removed only for appearance because it does not attack magnesium. Complex welded joints can be made with this flux because thorough cleaning is unnecessary.

Magnesium alloys are used in many forms in the aircraft, aircraft engine and commercial vehicle industries. Recently these alloys have extended to the textile and printing fields.



ASSEMBLING one of the largest spherical roller bearings ever made in the United States, to be installed in a Wisconsin paper mill.

Bearing Selection

Consideration of anti-friction bearing life and its relationship to load.

BRYCE T. RULEY
Senior Engineer, SKF Industries, Inc.

ACCURATE CALCULATIONS of the required size of an anti-friction bearing depends upon careful definition of several concepts.

If a number of identical bearings are tested to fatigue failure under identical loads, a considerable dispersion in individual lives is found, as Fig. 1 illustrates. The "life," L_x, for a group of bearings is defined as the number of revolutions that 90 percent of the group can be expected to equal or exceed before failure under the given identical loads. This life is often called the "design life" or the "10 percent failure life." It is taken as a basis for choosing bearings to insure high reliability in bearing use. The average life, L_m, of the group of bearings is about 5 times the design life.

From a number of tests it has been found that the design life of a group of bearings varies inversely as the cube of the load. If one group of bearings is tested under a load F_1 and gives a life L_1 while a second group of similar bearings is tested under a load F_2 and gives a life L_2 then

$$\frac{L_1}{L_2} = \left(\frac{F_2}{F_1}\right)^3 \qquad (1)$$

The "specific dynamic capacity" is defined as the pure radial load C that gives a design life of one million revolutions when the inner ring of the bearing rotates relative to the direction of load and the outer ring is stationary relative to the direction of load. Then for any radial load P the life is given in millions of revolutions by

$$L = \left(\frac{C}{P}\right)^3 \qquad (2)$$

If the value of C is known for a

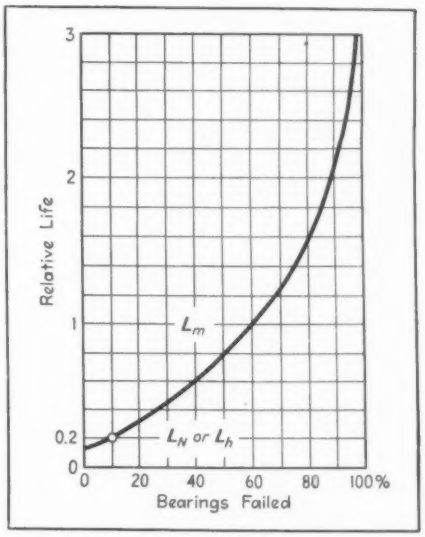


FIG. 1—A typical life distribution curve showing the relative lives at which various percentages of the bearings will fail under identical load. L_m is average life, L_N or L_h is design life or "10 percent failure life."

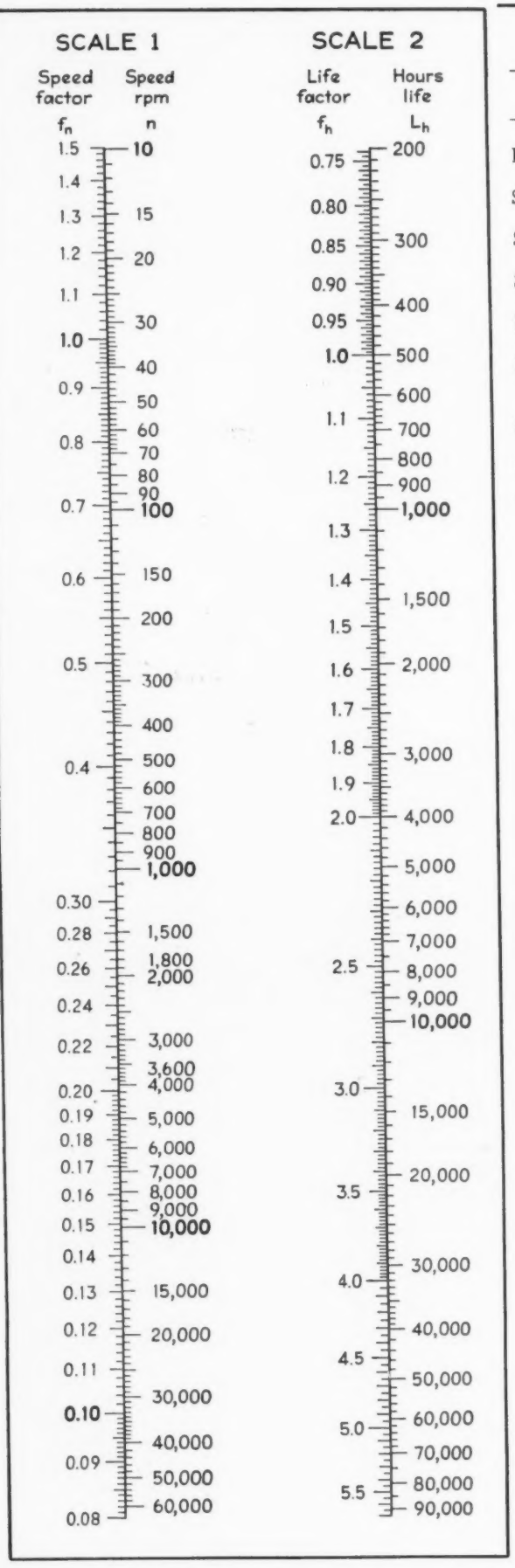


FIG. 2—Speed and life factors for any desired speed or life can be selected from their respective scales for substitution into one of the equation (8) formulas.

Table I—Load Factor

Bearing Type	f_c
Double row self-aligning ball bearings	3,200
Single row deep groove ball bearings without filling slot	6,400
Single row deep groove ball bearings with filling slot	5,300
Single row angular contact ball bearings	6,400
Double row angular contact ball bearings	5,700
Radial roller bearings with short rollers ($l_w < 1.4\ D_w$) and guiding flanges	7,100
Single row thrust ball bearings	8,500
Spherical roller thrust bearings	17,800

Table II—Rotation and Thrust Factors for Determining Equivalent Radial Load

Bearing Type		Rotation Factor X		Thrust Factor Y
		Rotating Inner Ring Load	Rotating Outer Ring Load	
Double row bearings	Self-aligning ball	1	1	$0.5 \cot \alpha$
	Angular contact ball, Spherical roller	1	1.4	$0.8 \cot \alpha$
Single row bearings	Angular contact ball, Tapered roller	0.5*	0.7*	$0.4 \cot \alpha$
	Deep groove ball bearings without filling slot	1	1.4	1.0–1.5
Cylindrical roller bearings		1	1.4	——

* However $P \geqq 2\ X\ F_r$

Table III—Thrust Factor

Bearing Type	Y_a for Load Rotating Relative to	
	Inner Ring	Outer Ring
Spherical roller thrust bearings	1.8	2.1

Table IV—Coefficients of Bearing Friction

Self-aligning ball bearings	$\mu = 0.0010$
Cylindrical roller bearings with short flange-guided rollers	$\mu = 0.0011$
Thrust ball bearings	$\mu = 0.0013$
Single row deep groove ball bearings	$\mu = 0.0015$
Tapered and spherical roller bearings with flange-guided rollers	$\mu = 0.0018$
Needle bearings	$\mu = 0.0045$

given bearing, the life can be estimated by the above formula for any radial load.

Some bearing catalogues list the value of C for each type and size of bearing. If C is not given it can be calculated from the following factors used in formulas below:

C = the specific dynamic capacity in pounds
i = the number of rows of rolling elements
D_w = the diameter of the rolling elements in inches
l_w = the length of the rolling element in inches
Z = the number of rolling elements per row
α = the angle of contact of the rolling elements with the outer race
f_c = the factor depending on type of bearing and conformity of contact. For some bearings of conventional type and groove radius, f_c can be found from Table I.

Expressed in the formulas

$$C = \frac{f_c\, i\, D_w^2\, Z^{2/3} \cos \alpha}{1 + 0.5\, D_w} \text{ lb} \qquad (3)$$

for ball bearings and

$$C = f_c\, i\, D_w\, l_w\, Z^{2/3} \cos \alpha \text{ lb} \qquad (4)$$

for roller bearings.

The angle α can usually be found with sufficient accuracy by scaling manufacturer's drawings. Deep-groove ball bearings and cylindrical roller bearings have $\alpha = 0$ degrees. Angular contact ball bearings are made in this country to $\alpha = 15, 20, 25, 30, 35, 40$ deg, depending on the manufacturer and the use for which the bearing is intended. Spherical roller bearings have $\alpha = 7$ to 15 deg and spherical roller thrust bearings have $\alpha = 50$ degrees. Tapered roller bearings have contact angles ranging from approximately 12 to 27 degrees.

Some bearings are designed solely or principally to carry thrust loads. These types include ball thrust bearings, spherical roller thrust bearings, tapered roller thrust bearings, and cylindrical roller thrust bearings. For thrust bearings the specific dynamic capacity C is given as a pure thrust capacity by the formulas

$$C = f_c \frac{D_w^2\, Z^{2/3}}{1 + 0.5\, D_w} \text{ lb} \qquad (5)$$

for ball thrust bearings and

$$C = f_c\, D_w\, l_w\, Z^{2/3} \sin \alpha \text{ lb} \qquad (6)$$

for single row thrust roller bearings.

Often the life of the bearing may be desired in hours at some definite speed in rpm of the shaft. Then a speed factor f_n and a life factor f_h can be taken from the scales in Fig. 2 and used in the following formulas:

$$P = \frac{f_n\, C}{f_h}, \quad C = \frac{f_h\, P}{f_n}, \quad f_h = \frac{f_n\, C}{P}, \qquad (7)$$

to find whichever quantity is unknown.

Many radial bearings are so constructed that they can carry both radial and thrust load. Here it is usual to calculate an equivalent radial load P, which would give the same life as the actual combined load. Where

P = the equivalent radial load in pounds
X = a rotation factor given in Table II
F_r = the radial component of load in pounds
Y = a thrust factor in Table II
F_a = the thrust component of load in pounds

Then P is found from the formula

$$P = X F_r + Y F_a \text{ lb} \qquad (8)$$

The factor X depends on whether the inner ring rotates relative to the direction of load, "rotating inner ring load," or whether the outer ring rotates relative to the direction of load, "rotating outer ring load." The factor Y depends on the type of bearing and the contact angle.

Most thrust bearings cannot carry any radial load. For those which can the equivalent thrust load is given by the formula

$$P_A = F_a + Y_a F_r \text{ lb} \qquad (9)$$

where Y_a is a factor given in Table III. The formula is valid only if the radial load is less than two-thirds of the thrust load at all times.

Friction

In some precision machines, consideration must be given to the friction in the bearing, and the additional torque required to overcome it. This friction varies with load and speed but its value can be approximated for normal operating conditions where

T = the torque in inch-pounds
d = the bearing bore in inches
P = the equivalent load in pounds (See equation 8 for a method of calculating P)
μ = the coefficient of friction as given in Table IV

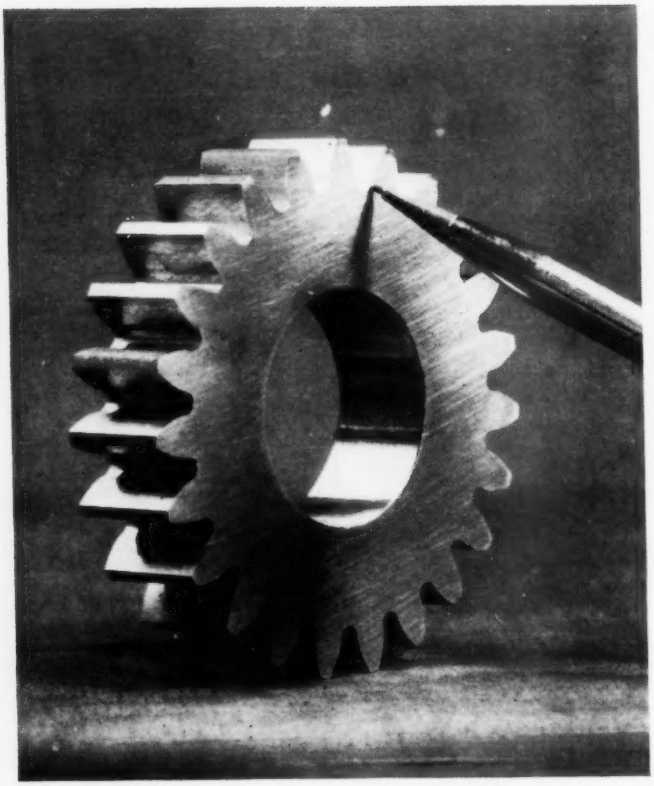
by

$$T = \frac{\mu d P}{2} \qquad (10)$$

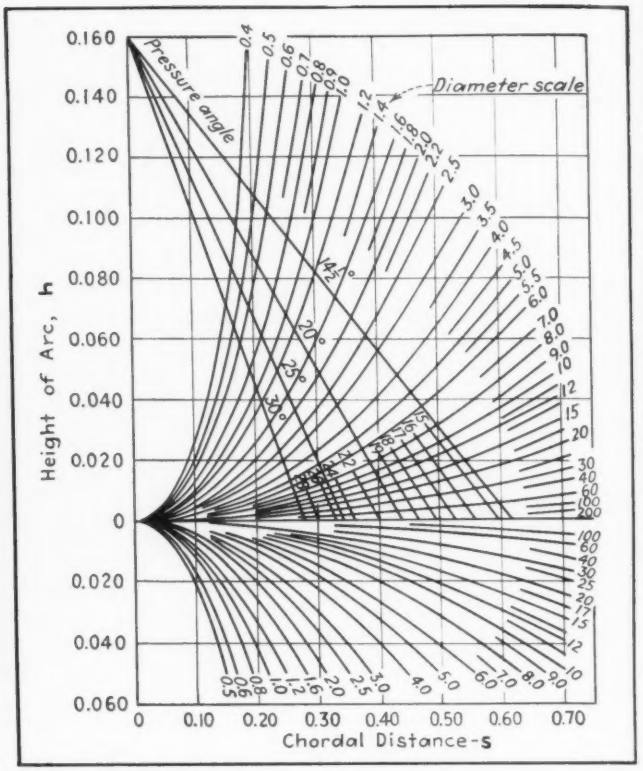
In addition to bearing friction, the friction of rubbing type seals must be considered where friction loss can cause trouble. Seal friction is normally much greater than bearing friction. Minimum seal friction can be obtained with nonrubbing labyrinth seals.

These considerations and good engineering judgment will serve as a guide to the specification of a bearing in ordinary applications. For unusual applications a complete analysis should be made by a qualified bearing engineer.

SPHERICAL ROLLER THRUST BEARING incorporating high load capacity with high speeds and low temperatures. Applications include wind tunnel fans, marine propeller shafts, large vertical water pumps, and electric generators.



UNDERCUT produced with preshave hob. A method for rapid solution of meshing problems is desirable for gear designers.



INVOLUTE ACTION CHART—1. This graph is used to solve meshing problems for diametral pitches in 6 to 12 range

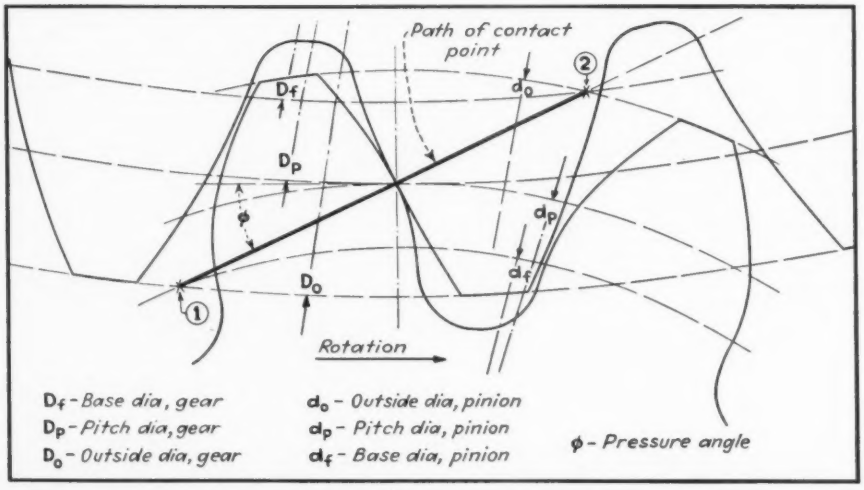
Graphical Determination of Gear Contact Ratios

D. W. DUDLEY

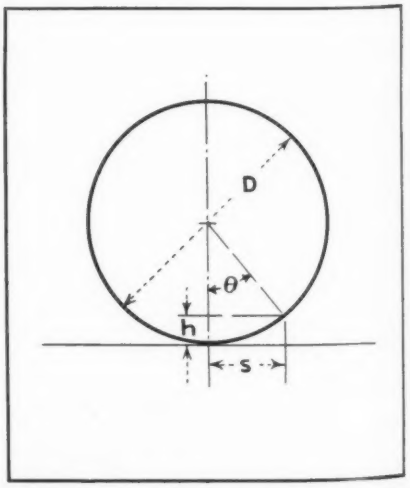
Gear Engineering Division
General Electric Co., Lynn, Mass.

Contact ratios, form diameters, hob undercut diameters and other gear action problems can be solved quickly and accurately with one type of curve chart. Preparation of the charts is explained and their use illustrated with sample meshing problems.

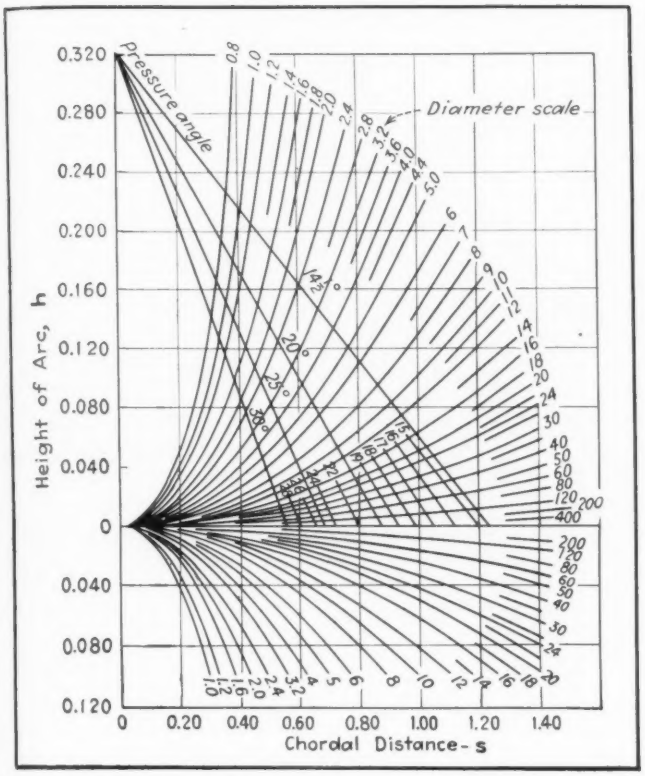
WHEN AN INVOLUTE GEAR and pinion roll together in mesh, it is necessary that their tooth profiles be conjugate curves to such a depth as will permit complete engagement without interference. If a continuous transmission of motion is to occur, the duration of contact between a pinion and a gear tooth must be sufficient to allow a second pair of teeth to come into engage-



MESHING DIAGRAM, showing that travel of contact point is predicated on intersection of circles and straight lines. Therefore graphical solution of meshing problems is possible.



POINT on circle is located in rectangular coordinates by chordal distance and height.



INVOLUTE ACTION CHART—2. for diametral pitches in 3 to 6 range. Both charts should be enlarged to 6 by 9½ inches.

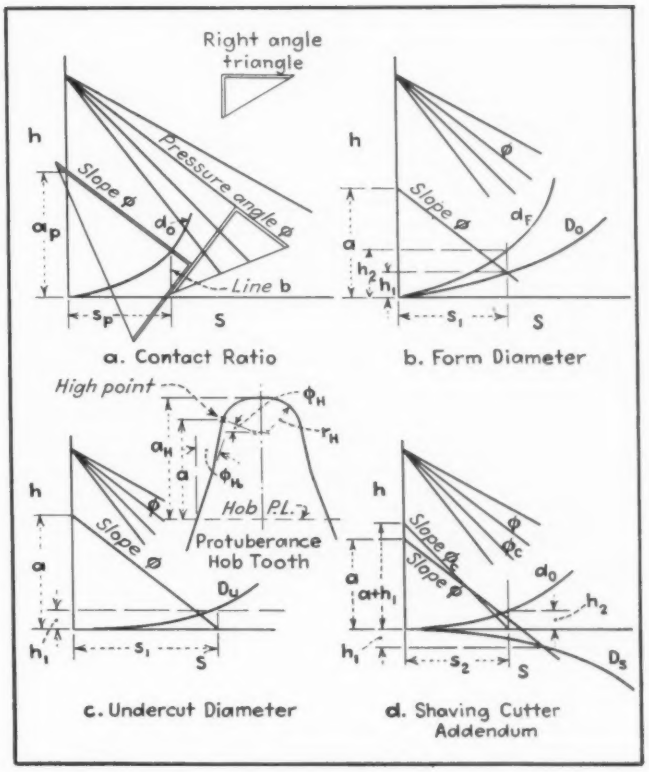


DIAGRAM OF METHODS of using the type of graph shown at the left to solve meshing problems, such as contact ratios.

ment before contact between the first pair is terminated.

These two requirements make it necessary to solve problems concerned with the limit or lowest point of involute action. These might be called "meshing diagram" problems, since they can be solved by constructing meshing diagrams of sufficient size and accuracy to permit measurement.

Tooth action of an involute gear and pinion is shown below, left. Contact starts at point 1 where the outside diameter of the gear cuts the line of action, and contact ceases at point 2 where the outside diameter of the pinion cuts the line of action. As the pinion revolves, the point of contact travels in a straight line between these points. Since both points are dependent only upon circles and straight lines, it is possible to find their positions graphically.

Shown to the right of the Meshing Diagram is a circle with a unit diameter and a point on the circle located in rectangular coordinates by a chordal distance s and a height h. These values may be calculated from a central angle θ by the equations,

$$s = \frac{D \sin \theta}{2} \qquad (1)$$

$$h = \frac{D\,(1 - \cos \theta)}{2}$$

Using equations (1), it is possible to assume a series of diameters and a series of angles θ and construct a graph sheet such as Involute Action Chart 1. Convenient scales can be selected so that any desired accuracy can be obtained. Table I shows sample calculations that can be used in plotting such a curve sheet. In these figures the pressure angle lines are drawn at a slope, h/s, which equals the tangent of the transverse pressure angle of the gear. Their intersection point at 0.160 in. on the h scale of Chart 1 has no particular significance. In the problems to follow, the pressure angle lines are used only to establish slopes; no readings are taken directly from them.

To solve gear problems in the 6 to 12 diametral pitch range, Chart 1 should be enlarged to 6 by 9½ in. approximately. Both Charts 1 and 2 have been reduced for reproduction here. By simply multiplying all linear values in the scales, it is possible to convert Chart 1 to another curve sheet suitable for doing gears of a different pitch range. Chart 2 is similar to Chart 1, except that the scale has been enlarged to handle 3 to 6 diametral pitch. Note that there is a 2 to 1 ratio between the two charts on the scales for h, s, and diameter. In a similar manner it is possible to convert the graph scales to accommodate any other desired diametral pitch range.

Contact Ratio

As a first problem, the contact ratio, m_p, will be determined. This quantity is often described as the "average number of teeth in contact." It is numerically equal to the length of the line of action, divided by the base pitch. Diagram a at the right of the charts shows how the graph of Chart 1 can be read to determine a_n sp dis-

Table I—Sample Calculations for Chart 1

Item	Operation				
1	Diameter, D	4 in.	4 in.	8 in.	8 in.
2	$D/2$	2 in.	2 in.	4 in.	4 in.
3	Central angle, θ	5 deg	10 deg	3 deg	6 deg
4	Sin θ	0.08716	0.17365	0.05234	0.10453
5	s = (2) × (4)	0.1743	0.3473	0.2094	0.4181
6	Cos θ	0.99619	0.98481	0.99863	0.99452
7	1−(6)	0.00381	0.01519	0.00137	0.00548
8	h = (2) × (7)	0.00762	0.03038	0.00548	0.02192

Table II—Steps in Determining the Contact Ratio

Item	Operation	Numerical Value
1	Pinion addendum, a_p	0.100
2	Pressure angle, ϕ	20 deg
3	Outside dia of pinion, d_o	3.200
4	Read chart 1 for s_p at (1), (2) and (3)	0.230
5	Gear addendum, a_G	0.100
6	Outside dia of gear, D_o	9.200
7	Read chart 1 for s_G at (5), (2) and (6)	0.255
8	(4) + (7)	0.485
9	Circular pitch, p_c	3.1416
10	Cosine of (2)	0.94
11	(10) × (10)	0.883
12	(9) × (11)	0.2775
13	m_p = (8) ÷ (11)	1.75

Table III—Steps in Determining the Form Diameter

Item	Operation	Numerical Value
1	Addendum of mating part, a	0.100
2	Outside dia of mating part, D_o	9.200
3	Transverse pressure angle, ϕ	20 deg
4	Read chart 1 for h_1 at (1), (2) and (3)	0.007
5	S_1 at (4)	0.255
6	Pitch diameter, D	3.000
7	2 × (1)	0.200
8	(6) − (7)	2.800
9	Read chart 1 for h_2 at (5) and (8)	0.0235
10	(9) + (4)	0.0305
11	2 × (10)	0.061
12	D_F = (8) + (11)	2.861
13	Check (9) using (12) instead of (8) and substitute new h_2 if a different value is obtained. Then correct (9), (10), (11) and (12) and recheck.	

Table IV—Steps in Determining the Undercut Diameter

Item	Operation	Numerical Value
1	Addendum of hob, a_H	0.135
2	Hob tip radius, r_H	0.038
3	Normal pressure angle hob, ϕ_H	19 deg 22 min 12 sec
4	Sine (3)	0.332
5	1 − (4)	0.668
6	(2) × (5)	0.0255
7	a = (1) − (6)	0.1095
8	Hobbing pitch dia of gear, D	3.000
9	2 × (7)	0.219
10	(8) − (9)	2.781
11	Hobbing transverse pressure angle gear, ϕ	20 deg
12	Read chart 1 for S at (7), (11), and infinite dia	0.300
13	Read chart 1 for h at (12) and (10)	0.0325
14	2 × (13)	0.065
15	D_u = (10) + (14)	2.846
16	Check (13) using (15) instead of (10). Revise (13), (14) and (15) if new h is obtained and recheck.	

Table V—Determining the Shaving Cutter Addendum

Item	Operation	Numerical Value
1	Shaving diameter, D_s	2.852
2	Shaving pitch dia of gear, D	3.000
3	(2) − (1)	0.148
4	(3) ÷ 2	0.074
5	Shaving transverse pressure angle of gear, ϕ	20 deg
6	Read chart 1 for h_1 at (4), (5), and (1) on lower diameter scale	0.0295
7	(4) + (6)	0.1035
8	Shaving pitch dia of cutter, d_C	7.372
9	2 × (7)	0.207
10	(8) + (9)	7.579
11	Shaving transverse pressure angle of cutter, ϕ_C	19° 32′ 46″
12	Read chart 1 for s, at (7), (11), and infinite dia	0.290
13	Read chart 1 for h_2 at (12) and (10)	0.0115
14	a_C = (7) + (13)	0.115
15	2 × (14)	0.230
16	d_o = (8) + (15)	7.602
17	Check (13) by substituting (16) for (10). If new h_2 is obtained revise operations (13) to (16) and recheck.	

tance for the pinion. The steps are as follows:

(a) Locate pinion addendum, a_p, on b scale.

(b) By using two right angle triangles, draw line parallel to pressure angle line to pass through a_p.

(c) Read s_p value at intersection of line (b) and curve for pinion outside diameter.

The steps (a), (b), and (c) are repeated using data for the gear to obtain an s_G value. Then the contact ratio can be obtained by the simple relation

$$m_p = \frac{s_p + s_G}{p_c \times \cos^2 \phi} \qquad (2)$$

Wherein,

p_c = circular pitch = 3.14159/diametral pitch
ϕ = pressure angle

Operation sheets listing each step of the problem make it possible to have the actual work done by calculators who may know little about gear engineering.

An operation sheet for the contact ratio problem is given in Table II. Nomenclature used agrees with the American Standard Letter Symbols for Gear Engineering. Numerical values are listed for a 30-tooth pinion and a 90-tooth gear, 10 diametral pitch, 20-deg pressure angle, and each having 0.100 addendum. Numerical values for the operations necessary to solve the problem are underlined in the table. A reader desiring to check the method should use the graphs and steps in Table II to solve for the underlined figures. It should take an experienced calculator about three minutes.

Form Diameter

The "theoretical involute form diameter, popularly known as "form diameter," is the lowest diameter on the tooth at which involute action occurs.

In the Meshing Diagram, the pinion form diameter is the diameter that passes through point 1. In precision gearing it is necessary to calculate this diameter precisely, and specify it on the production drawing so that involute profile checks can be taken from this diameter out to the tip of the tooth.

How to read the graph in solving for the form diameter is shown in Diagram b. The addendum and outside diameter of the mating part are used to locate the point at which involute action terminates. An h_1 value is read to find the amount by which the mating part has separated or pulled out of mesh. Then an h_2 value is read at the form diameter for the amount that the form diameter itself has pulled out.

Since the form diameter is not known precisely at this stage, it is necessary to read h_2 using an approximate value for the form diameter. Then h_2 can be corrected later if the assumed form diameter is substantially in error. In practice it is seldom necessary to do this.

Formula for the form diameter is,

$$D_f = D - 2a + 2(h_1 + h_2) \qquad (3)$$

Wherein,

D = pitch diameter of part
a = addendum of mating part
h_1 = pullout of outside diameter of mating part
h_2 = pullout of form diameter

Table III shows an operation sheet for determining form diameters. Numerical values are shown to set up the problem of finding the form diameter of the same 30-tooth pinion used in the first problem. Answers are underlined.

Undercut Diameter

Suppose that it is desired to hob gears with preshave hobs and then finish by shaving. The radius of the

hob tooth should be large enough to generate a generous fillet at the base of the gear tooth and thereby relieve stress concentration, and yet it should not be so large that the fillet will extend above the form diameter. The tool designer needs to design his hob so that the deepest point of undercut will come at or below the gear form diameter. This leads to the problem of determining the undercut diameter, D_u. In the photograph on the first page is illustrated a pinion tooth that has been hobbed with a "protuberance" type of preshave hob. The undercut produced by the hob is clearly shown. The undercut diameter passes through the point of tangency between the undercut and an involute curve.

The nature of the hob tip and the way to use the graph to solve this problem are shown in Diagram c. With single thread hobs the pullout of the outside diameter of the hob is usually negligible, so the problem can be solved by simply finding the hob addendum distance, a, to the highest point on the protuberance and the pullout of the undercut diameter. The formula for D_u is,

$$D_u = D - 2\,[a_H - r_H (1 - \sin \phi_H)] + 2\,h_1 \qquad (4)$$

Wherein,

D = hobbing pitch diameter of gear
a_H = hob addendum
r_H = hob tip radius
ϕ_H = normal pressure angle of hob
h_1 = pullout of undercut diameter

The operation sheet for determining the hob undercut diameter is in Table IV. Numerical given values are listed for the 30-tooth pinion with answers underlined. It is assumed that the pinion has 15-deg helix and a normal pressure angle of 19 deg 22 min 12 seconds. If the pinion had been shaper-cut instead of hobbed, the procedure for finding D_u would have been very similar to that of finding the form diameter. Table III could have been used, except that the addendum and outside diameter of the mating part would have had to have been the values of the highest point of protuberance on the shaper cutter and the pitch diameter, D, would have to be the actual operating pitch diameter in shaping.

Shaving Cutter Design

After the tool designer has designed his hob so that it will put the undercut diameter below the form diameter, he still has the problem of designing a shaving cutter that will make the diameter at which shaving ends come below the form diameter but not below the undercut diameter.

The shaving cutter problem is usually carried out in a backwards direction from the proceeding problems. Instead of solving for D_s (diameter at which shaving ends) from some predetermined cutter design, the desired value of D_s is already known and the calculation is carried out to determine what addendum the cutter should have to make it finish the involute to the proper depth.

The basic steps in reading the graph to determine the cutter addendum, a_C, are shown in Diagram d. Note that it is necessary to use two different pressure angles; namely, the transverse pressure angle of the gear and the transverse pressure angle of the cutter. Since shaving cutters nominally work on crossed axes, these values are not the same. Also it is necessary to read the graph of Chart 1 in the lower region where the circles are curving down instead of up.

The formula for the cutter addendum is as follows:

$$a_C = \frac{D - D_s}{2} + h_1 + h_2 \qquad (5)$$

Wherein,

D = pitch diameter of gear during shaving
D_s = diameter where last point of involute is shaved
h_1 = pullout of D
h_2 = pullout of cutter outside diameter

An operation sheet for the shaving cutter addendum problem is shown in Table V.

Numerical values are shown for the 30-tooth pinion, assuming that it has a 15-deg helix angle and that the shaving cutter has an 8-deg helix of the same hand, making the crossed-axes angle 7 degrees. With a 20-deg transverse pressure angle on the gear, the normal pressure angle will be 19 deg 22 min 12 sec for both the cutter and the pinion, and the transverse angle of the cutter will be 19 deg 32 min 46 seconds.

Limitations of Graphical Method

It must be kept in mind that the solutions are based on *involute* action. If the action occurs below the base circle, then true involute action is impossible and misleading results will be obtained by following the methods described. If the following equation gives a number greater than the number of teeth in the part, there is danger of noninvolute action. Enveloping layouts should be made to analyze the tooth action by showing positions of each hob tooth as they cut on the gear teeth or successive contacting positions of a mating gear tooth.

$$N = \frac{2\,a}{\sin^2 \phi} \qquad (6)$$

Wherein,

a = addendum of mating part or "generating" addendum of tool
ϕ = transverse pressure angle

Another limitation is the fact that hobs have finite numbers of teeth. Sometimes it happens that the teeth are so few on the hob that "steps" are formed instead of a continuous curve in the region of the hob undercut diameter. If the number of hob gashes divided by the number hob threads is less than 10, there is apt to be trouble with steps.

In calculating shaving cutters, it is necessary to keep the cutter addendum small enough so that the sum of the cutter and gear addenda do not exceed a certain percentage of the gear whole depth. For the examples shown $a_G + a_C$ must not equal more than 93 percent of gear whole depth. If this percentage becomes too large, the cutter will notch or foul the root fillet of the gear, even though its contact with the involute profile is satisfactory. The remedy is to reduce the cutter outside diameter and tooth thickness so that the cutter shaves at a pitch diameter which is less than the operating pitch diameter of the gear. In all of the above problems, the pitch diameters, pressure angles, and addenda must be the actual operating values rather than the design or theoretical values. For instance, a gear may be designed for 20-deg pressure angle, but it might be hobbed at a 19-deg hobbing pressure angle and it might be shaved at a 16-deg shaving angle. Where an "operating" pitch line is used that is different than the design pitch line, the cutting tool must have its tooth thickness adjusted so that it equals the space thickness of the gear at the operating pitch line. The normal pitch and normal pressure angle of the tool must agree with those of the gear at the operating pitch line.

The above 19-deg hobbing would make it necessary to locate the "hobbing pitch diameter," i.e. the diameter at which the involute vector angle was 19 deg, before proceeding with the calculation of the hob undercut diameter. Likewise, it would be necessary to locate the diameters corresponding to a 16-deg transverse pressure angle on the gear before the shaving calculation could be carried out.

If there was need for extreme accuracy, the Chart 1 curves could be plotted on a much larger piece of paper. In many cases in the past, calculations or layouts for the limits of involute action were not done because of the trouble involved in getting precise answers quickly. With the new method it is possible to have precise answers for every job.

Aircraft Design Superiority Recommended As National Policy

CONTINUING RESEARCH to maintain control of the air was strongly urged on the nation in a recently released report by a five-man commission appointed by President Truman. Titled "Survival in the Air Age", the report (see Washington Notes, page 159) outlined steps necessary to maintain air-superiority in a world that could, according to the report, see an atomic war at any time after "A-day", January 1, 1953.

Headed by Thomas K. Finletter, this commission has evaluated the strength of our present air force and made recommendations for a policy to be followed during the two phases of the future. Phase I covers the period to A-day and Phase II, after A-day. Appropriations, research, development and production programs must provide 12,400 modern planes and 8,100 reserve aircraft, to replace initial losses in the event of war, by the conclusions of Phase I.

CURRENT REQUIREMENTS

Need for intensive research and development in aeronautics is now felt because wartime efforts were aimed at developing for production existing designs of aircraft. Fundamental research was practically abandoned and by VJ-Day our reserve of research information was largely exhausted. Expert opinion holds that our Air Force must have planes that apply results of the latest fundamental investigation.

Research into fundamental aerodynamic phenomena in all speed ranges, particularly in the supersonic, should be pursued; as such speeds are important in design of high speed piloted aircraft and long-range guided missiles. Also, our knowledge of theory and application in accurate guidance of missiles to selected targets is deficient.

Most pressing current requirement is to establish an Air Force policy and set necessary plans and programs in motion. The Joint Chiefs of Staff will have to evaluate all suggestions relating to the Military Establishment, as outlined in the National Security Act of 1947, and determine priorities of the various plans.

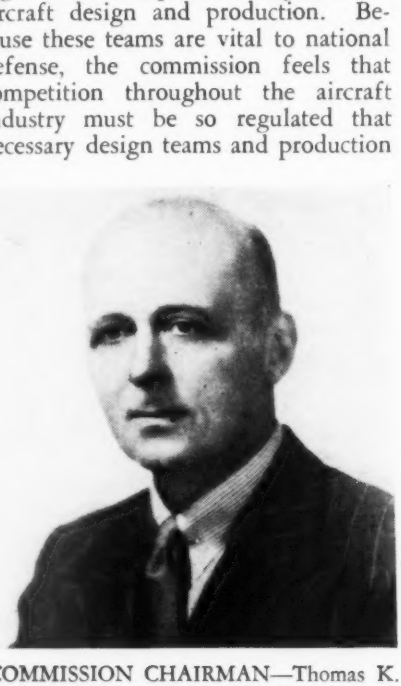
For fiscal year 1948, the government is spending about 312 million dollars for aeronautical research and development. This represents direct expenditures for aeronautical research, but is supplemented by expenditures in other contributing fields. These expenditures are broken down by agencies in the accompanying table.

Of the total appropriations to the services, about 200 million dollars go for research and development work done for the military by the aircraft industry and in educational and scientific institutions. The balance is spent in planning and evaluation by services in their own research and development facilities.

PERSONNEL PROBLEMS

As in many fields today, shortage of trained personnel is a serious problem in aeronautical research laboratories. The commission recognized this need and recommended that every encouragement be given universities and scientific institutions to train more and better aeronautical scientists, researchers and engineers in all fields.

During wartime, design teams were organized to expedite all phases of aircraft design and production. Because these teams are vital to national defense, the commission feels that competition throughout the aircraft industry must be so regulated that necessary design teams and production organizations are not liquidated. Design teams can also be eliminated if aircraft output falls too low. Procurement and production planning must be done so that the aircraft industry will not be forced to lay off these key teams.



COMMISSION CHAIRMAN—Thomas K. Finletter, prominent New York and Pennsylvania lawyer, served as special assistant to the Secretary of State between 1941 and 1944, and as consultant on international administration to the American United Nations delegation at San Francisco in 1945. Later this year Mr. Finletter will serve as chairman of the planning board for the International Air Exposition in New York.

The Air Policy Commission disagrees with the Attorney General's recent proposal that all rights to patentable inventions made under government-financed contracts be assigned to the government. Adoption of such a policy would turn research and development men's efforts away from government development to commercial activities.

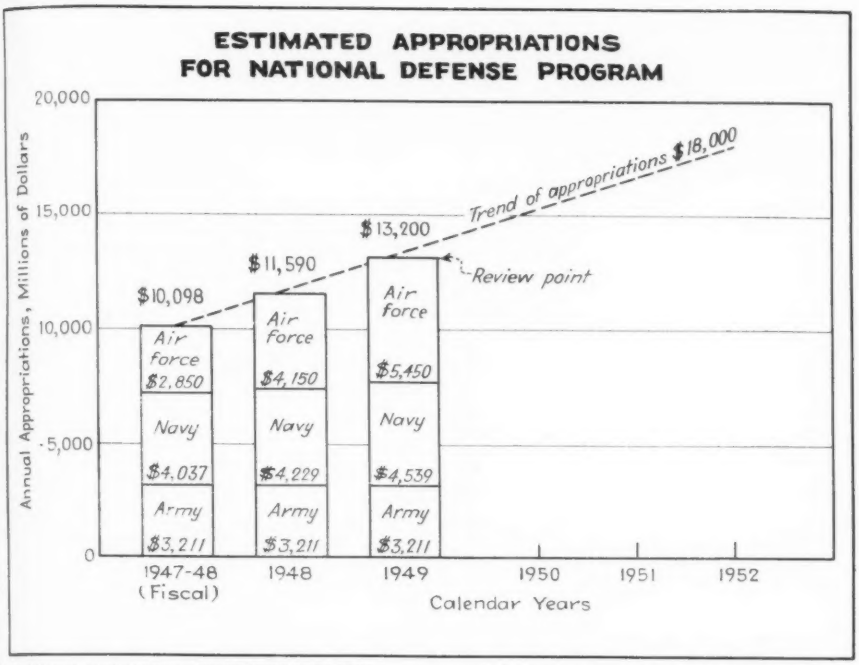
Dispersion of aircraft and parts plants, a desirable plan in wartime, could introduce personnel problems. Desirable personnel might not want to change job locations from well-developed areas to under-developed regions.

Another personnel problem is found in the services themselves. Present policies are devised to produce well-rounded officers. An officer's duties are regularly changed so he will have experience in all branches of the military. In long-range research planning, key personnel cannot be changed without losing continuity of effort and experience. The commission recommends that the services inaugurate plans for having a force of career officers in research activities.

CONTINUOUS RESEARCH

To maintain our Air Force in proper state of modernization, continuity of research is necessary. Plans should be established whereby each company will always have at least one aircraft in production, one in development and one in design-study stage. To successfully carry out this schedule, all manufacturers of airplanes and parts must have progressive research programs. Aircraft designs developed must be technically superior and readily producible. An airplane that can be produced easily can be maintained easily.

To increase research capacity during the war, universities and engineering schools were awarded government contracts to pursue research projects in aeronautics. Such projects answered fundamental questions and developed a nucleus of trained research



ARMED FORCES APPROPRIATIONS recommended by the President's Air Policy Commission showing continuous increase in annual Air Force expenditures up to "A-Day."

Aeronautical Research and Development Expenditures By all Governmental Agencies During Fiscal Year 1948

Agency	Expenditures	
	Amount	Percent of Total
Air Force	$145,316,000	46.6
Bureau of Aeronautics—Navy	75,000,000	24.0
National Advisory Committee for Aeronautics	43,449,000	13.9
Bureau of Ordnance—Navy	30,000,000	9.7
Ordnance Department—Army	11,000,000	3.5
Office of Naval Research	4,952,000	1.6
Civil Aeronautics Administration	1,670,000	0.5
Weather Bureau	521,000	0.2
Bureau of Standards (included in above items)	(838,200)	....
Total	$311,908,000	100.0

personnel. Because these services can continue during peacetime, the commission feels that they should be energetically supported.

Projects deserving continuous research efforts have been listed by the Air Policy Commission (not in order of importance):

ATOMIC PROPULSION. Possible use of atomic energy for the propulsion of planes and missiles should be investigated under supervision of the Atomic Energy Commission, Air Force, Navy and National Advisory Committee for Aeronautics. All background information on atomic knowledge must be made available to the agencies involved.

ELECTRONICS. Because of the rapid progress in electronic devices, the commission recommends formation of a better coordinating plan than now exists. Specific electronic applications requiring continuous research are: Detection of approaching aircraft or missiles; guidance of pilotless craft and missiles, and navigation and blind landing techniques for all aircraft.

GUIDED MISSILES. Techniques for producing successful short-range guided missiles are now understood, but problems of long-range missiles have not been surmounted. It is the commission's opinion that both subsonic and supersonic missiles should be completely investigated.

HELICOPTERS. Direct-lift, rotary-wing aircraft appear so promising that continuous research effort is warranted. Used as special auxiliary craft, helicopters have proven their worth.

BLIMPS. Nonrigid aircraft will be useful for carrying submarine detection devices of the future, as they were during the past war. The commission recommends continued research in this field to be carried out by the Navy and solely for naval applications.

POWER PLANTS. Though gas turbines and rocket engines are expected to replace aircraft reciprocating engines, the commission feels research on these types should not be dropped. Reciprocating engine and propeller combinations will be used for many years on long-range bombers and transports. High fuel consumption and short service life of jet engines are projects that must be solved as rapidly as possible.

GENERAL. The President's Air Policy Commission agrees with military opinion that rigid aircraft have little military use, and does not advocate government sponsorship of airship research. Personal aircraft, military "grasshoppers" during the past war, should be developed to a high degree. NACA research into small craft flight principles should not extend to development stages. If basic theory is discovered, industry will do the necessary developing. Air navigation and blind landing methods are fields that will return immediate results in commercial aviation and long-term benefits in military operations.

The commission's policy objective is to provide a modern air force, backed by projects ready for production at a moment's notice.

RECOMMENDATIONS AID INDUSTRY

Recognizing the aircraft industry as vital to national defense, the commission made recommendations that would stabilize the industry and maintain a producing nucleus of aircraft builders, suppliers and designers.

It was recommended that the company developing an airplane should be given the contract for its construction, where this is possible. Production airplanes differ materially from original design, as service requirements change, and developing company can make these changes most rapidly.

Also it was acknowledged that research programs cannot be laid out cost-wise previous to experimentation. For this reason, the commission recommended that appropriations to aeronautical research agencies be estimated on past performance of the agency and importance of the project. Funds should be allocated as a lump sum, and the agency should have blanket permission to distribute funds to meet requirements of various projects on its program.

Forward contract authorization and long-range planning by the armed

services would materially aid aircraft companies. The commission believes that military aircraft production should be planned five years in advance and reviewed each year. Contracts should be written with unit deliveries spread over the five-year period. This would allow aircraft companies to order materials in large quantities and would keep the builders and suppliers in continuous operation. Savings could amount to 20 or 25 percent.

General Policies

The National Security Act of 1947 vests responsibility for coordination of military, industrial and civilian mobilization in the National Security Resources Board. The commission recommends that Under Secretaries of Army, Navy and Air Force give special attention, under this board, to industrial mobilization—attention comparable to that given research, development and procurement. Prior planning is essential, as stated in the report, and all controls and plans should be geared to start immediately upon the declaration of an emergency by the President.

The Air Coordinating Committee proposed that a reserve of general purpose machine tools be established and maintained with a minimum of 65,000 units. This program has started, and the President's Air Policy Commission agrees that it should be carried to completion.

Most general of the commission's policy recommendations concerns release of aeronautical information. At present, physical dimensions, performance characteristics, technical data and clear photographs are released on most aircraft designs. The commission believes wartime security regulations should be imposed and rigidly enforced on all aeronautical discoveries. However, also contrary to present policy according to the report, we should publicize our military readiness and potential strength.

Throughout its investigation and report, the President's Air Policy Commission concentrated on aviation and made no effort to evaluate overall military necessity. This task has been assigned to the Joint Chiefs of Staff, whose responsibility it is to evaluate plans and assign priorities.

Nondestructive Thickness Gaging by X-Ray Successfully Applied

X-ray thickness gaging enabling continuous production of accurate sheet and strip is now possible with a recently announced Westinghouse Electric Corporation instrument. This will be used in measuring thickness of all types of material, including steel, possibly coated with lacquer; glass, plastics and nonferrous metals. The gage automatically rejects off-gage material in flying shear applications or controls roller screw downs to maintain uniform thickness in material coming from a cold mill.

Basic advantages of an X-ray thickness gage include: No contact with material, surface will not be marred, thin foil will not be torn, and no part of the gage will be worn; instantaneous response to changes, and ready access to areas difficult to gage by other methods.

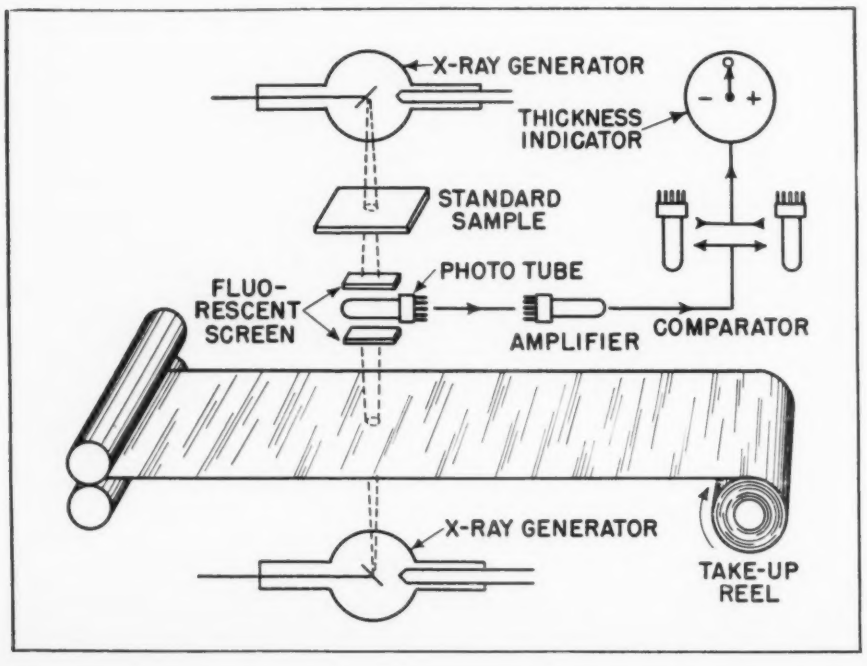
Gaging is accomplished by comparing amounts of X-ray absorption by an unknown piece with absorption by a piece of same material with desired thickness. Two X-ray sources (see schematic drawing) are focused on a photo tube pick-up so that the pick-up is at the right angle formed between the ray paths.

Radiation from the lower source is directed up through the sheet being gaged so that transmitted rays will strike and illuminate the fluorescent screen of the pick-up. X-rays from the upper source are directed horizontally through a standard sample of correct thickness. The two sources emit radiation 180 deg out of phase so their waves alternate in striking the pick-up. This tube compares the intensities of the two X-ray beams after a large part of the radiation has been absorbed in the standard sample and the test sheet.

The pick-up device is a photomultiplier tube surrounded by a fluorescent screen and inclosed in a lightproof cover. A photomultiplier tube is a phototube with a built-in electron amplifier—permitting use of low-powered x-ray generators. Absorption of x-radiation in the fluorescent screen excites the compound of the screen to visible luminescence which is picked up by the photo electric cell and converted into current pulses. The latter vary in amplitude according to the thickness of the material.

Variations in X-ray intensity are registered on the indicator instrument. At full deflection of indicator needle, a red light goes on and a throw-out, or other mechanism, is actuated through electronic circuits. One model of the gage has been designed for thickness of steel sheet from 0.005 to 0.050 in. The gage has a full scale sensitivity of one percent and will indicate a deviation of 0.00001 in. in a 0.01 in. sheet.

The basic design is being modified for inspection of complicated shapes; this is expected to be applicable to production of bottles and similar objects with internal voids.



Washington Notes

Federal Research Expenditures Grow Larger and Larger

Active federal support of research and development shows signs of assuming major proportions in the near future. Recent presidential reports emphasized the urgent need for government aid to science. For example, President Truman's Economic Report to Congress said:

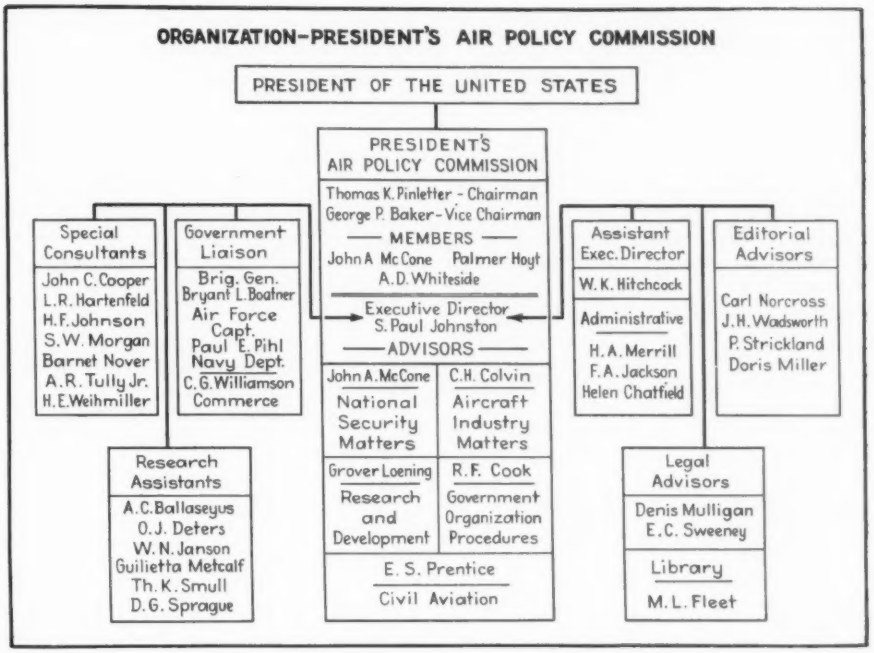
"The expansion of national output, as well as national security, depends increasingly upon scientific accomplishments. Before the war we drew heavily on the basic discoveries of scholars all over the world. The war has impeded this free flow of knowledge . . . and destroyed much of it at the source . . . This is need for enlarged research . . . " This statement finds support in the President's budget for the fiscal year 1949.

Proposed federal expenditures for research and development by all governmental activities for 1949 show an increase of $57,000,000 over the current year, from $884,000,000 to $941,000,000. This seems large; however, consider that the cost of researching and developing only one aircraft, the Consolidated Vultee B-36, was $30,000,000. This figure assumes only the building of the first prototype, and does not take into account the tremendous added expense of developing a production model.

Close perusal of the 6 lb, 1,353 page Budget of the United States shows that a minimum of 500 million dollars is requested for research and development in fields of interest to engineers. This figure does not include the payment of personnel or construction of facilities. With funds requested for these two functions included, the total bill to the taxpayer would run well over a billion dollars.

This 500 million dollar figure is the amount of money the President requests be made available for research and development obligations, in addition to current contracts and former appropriations. The actual money to be spent in fiscal 1949 falls in the category of expenditures and is not available except in the over-all figure —941 million dollars.

Continuing Air Research—The report of the President's Air Commission, "Survival in the Air Age" (see article, page 156, and accompanying chart) urges government-aided expansion in every conceivable phase of aeronautics. It calls for the maintenance of more than a token air force, and, more importantly, insists that this air force, with a large and adequate reserve, be kept constantly up to date. The report gives us until 1953 to get this done, saying that after that, we will no longer have a monopoly on atomic power and weapons.



This report is important because the chances are that it will become the guide for future aeronautical development.

The report points out that "by VJ Day our reserve of research information was largely exhausted." It emphasizes that development can proceed no faster than fundamental research. It urges that research projects not be limited by budgetary restrictions, that no fixed amount of money should be allocated to any one item of study, but that the agency should have the right of discretion in distributing funds to its various projects as demand arise. In order to meet emergencies arising during the fiscal year, the report advises that each agency be alloted annually a revolving fund for the construction of new facilities. At the end of the year the agency would be required to make an accounting of its disposition of the money.

The commission urges that the present 2-year limit set on contracts be eliminated, pointing out that many projects at the outset are known to be long term problems. In order to encourage manufacturers to more readily accept research and development contracts the report suggests a liberalizing of government cost allowances.

NACA is recommended as the logical agent to coordinate all research in the aeronautical sciences. The report states that NACA should be granted additional funds to strengthen its organization, in order that it can better fulfill this purpose.

Air Appropriations—Aeronautics and associated fields account for the largest part of this research and development money. Research includes everything from the ceramics of jet turbine blades to "atmospheric" conditions in outer space.

Air Force funds available for research and development obligations for 1949 are almost the same as 1948 —145 million dollars. This is partially explained by the fact that much Air Force research activity is actually being conducted under "the procurement of planes" (767 million dollars for 1949), "the procurement of guided missiles" ($10,300,000), or under other titles where detection is intentionally difficult.

Additional research funds requested for the Army are as follows: $6,246,000 for the Quartermaster Corps, including the study of everything from tropical food containers to "Operation Frostbite" clothing; $18,350,000 for the Signal Corps, encompassing products such as airborne wire laying reels and special antennae; $5,900,000 for the Corps of Engineers; $40,567,900 for the Ordnance Department, and $6,000,000 for the Chemical Corps.

Naval research and development chiefs are asking for $210,831,532

for 1949. The biggest single slice is for aeronautics, 75 million dollars. The problems of jet propelled fighter operation on carriers will get careful attention along with the Navy's extensive transonic and supersonic projects.

The Office of Naval Research, conducting basic work in chemistry, physics, nucleonics, electronics, metallurgy, geophysics, and others, sets its needs at 40 million dollars. In addition to conducting basic research and supporting the Naval Research Laboratories, ONR also operates a Special Devices Center on Long Island, where aerial gunnery and navigation trainers are developed.

Research and development in the various Navy agencies is closely interlocked. For instance, Navy Ordnance which is earmarked for $63,000,000, supports and operates the new Naval Ordnance Laboratory at White Oak, Md. Most of NOL's work will be on projectiles and explosives, but much will be learned from guided missile research in these tunnels which will be directly applicable to supersonic aerodynamics.

The balance of Navy research money requested is as follows: Bureau of Ships, $30,350,000, and the Bureau of Yards and Docks, $2,481,532.

The needs of the Atomic Energy Commission for funds to be devoted specifically to research call for $70,394,060. This does not include attendant construction and personnel salaries. AEC wants $338,481,808 for construction purposes alone in 1949, much of which will be devoted to building research facilities.

As in fiscal 1948, the budget again assumes the formation of a National Science Foundation, and includes a request for 15 million dollars for this purpose. Congress passed a bill for its formation this last year, but the President, objecting to the way in which it was to be administered, disapproved it.

The rest of the half billion dollars requested for research and development is wanted for TVA, mining and fuel research of the Department of the Interior, NACA, the Bureau of Standards, the Civil Aeronautics Administration, and other Department of Commerce activities.

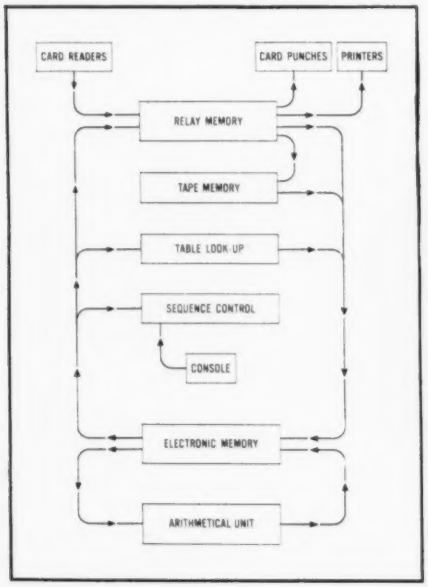
Huge Electronic Calculator to Solve Industrial and Scientific Problems

OFFERING A NEW TOOL FOR INDUSTRY, science and business, a new high-speed calculating machine, the IBM Selective Sequence Electronic Calculator, was dedicated recently by the International Business Machines Corporation, at its New York headquarters. Based on an extension of principles learned through development of ENIAC, the original electronic calculator (see PRODUCT ENGINEERING, May, 1946, page 412), the new machine has already achieved worthwhile results in such diversified fields as astronomy and gear design.

Containing 12,500 electronic tubes, the new machine combines the speed of electronic circuits with a memory capacity of 400,000 digits and the necessary controls to utilize this speed and capacity on the most complex problems of science. The new calculator combines electronic speed, vast memory capacity and highly flexible and convenient programming (or sequencing) facilities. Its flexibility exceeds that of the IBM Automatic Sequence Controlled Calculator presented to Harvard University in 1944, and its over-all productive capacity is far in excess of anything previously achieved.

In many respects, the IBM Selective Sequence Electronic Calculator follows the pattern of man's mind in performing sequences of complex calculations. By means of a "central nervous system," the program devised by the scientist for solution of a problem automatically directs the sequence of operations, selects the proper numbers from the various memory units or from the reference tables, directs them to the calculating unit, guides the calculating processes and routes the results back to their proper places in the memory unit. When the desired results are obtained, the program directs the machine to record them.

The high speed of calculation in this new machine is attained by electronic circuits for computing and control. All the fundamental arithmetical operations that may be called



(Left) Schematic diagram shows number paths in IBM Selective Sequence Electronic Calculator. Units are not located in this physical order in actual machine. (Right) General view—exterior styled in collaboration with Sunberg-Ferar, Detroit industrial designers.

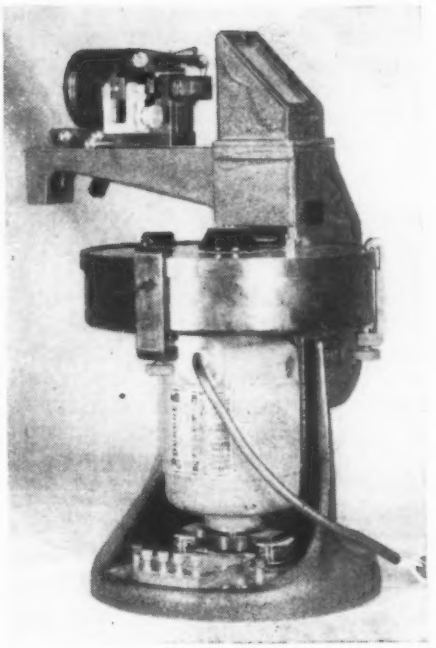
for in the processes of higher mathematics are performed electronically. Additions, subtractions, multiplications, divisions, shifting columnar relationships of numbers, and rounding off results to the required number of digits are all performed without the use of moving parts.

The great masses of numerical data and instructions used in complicated problems are fed into the machine automatically at high speed. Facilities are provided for assembling this material automatically from various sources within the machine and for sorting, checking and arranging it in the most efficient order for solution of the problems.

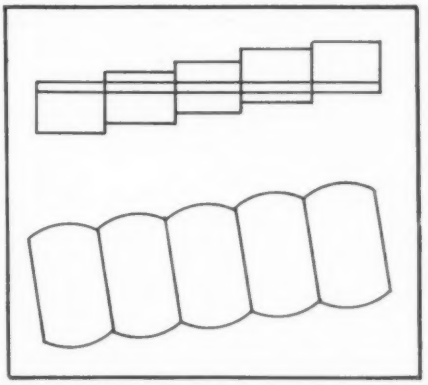
The memory capacity far exceeds that of any other calculating machine —a total of nearly half a million digits. It can remember this number of digits and recall them automatically as required. Those to be recalled most quickly are held in electronic circuits, and the remainder, recoverable as rapidly as they are needed, are stored in relays and as holes in continuous cardstock tapes. And by using punched cards as a supplementary medium of storage, the memory capacity is made almost limitless.

Programming facilities have been made so automatic that the operator need give no more than general instruction to carry out comparatively long sequences of operations.

Results can be recorded in either or both of two ways at high speed. One is in the form of printed records which are obtained at the rate of 24,000 digits a minute. The scientist can follow the progress of a computation and modify his program in the light of the results as they are derived. This will be of particular advantage because this new calculator will be devoted largely to research problems in which calculations cannot be completely planned in advance. The second means of recording results is in the form of punched holes in cards for automatic use at a later time either by the calculator itself or by other devices. For example, during a calculation it is not necessary to decide what results are to be saved or published, or in what form. Provision is made for later automatic printing from selected machine records.



High-speed camera showing dissecting lenses at top and film rotating drum at center. Drum mounted on driving motor shaft.



Multiple lens system for dissecting picture image showing (below) inclination of lenses and (above) strip position across lens images in the world's fastest camera.

Fastest Camera Aids High Speed Studies

TEN TIMES FASTER than any camera previously made, a shutterless motion picture camera that takes pictures at rate of 11 million per second has recently been designed by Dr. Brian O'Brien, Director, and Gordon G. Milne, Research Associate, the Rochester Institute of Optics, University of Rochester.

When a motion picture of a rifle bullet, taken during a flash of light lasting 1/5,000 sec, is projected on a screen at the usual speed, it requires 1 min to show 1 in. of movement.

Exposure by conventional means at speeds of several million frames per second would require film movement through camera at 20 to 50 times bullet velocities. If, instead, film is stationary and image is caused to move by rotating mirrors, total number of frames and total duration of motion picture become limited.

These difficulties are avoided if the image of the rectangular picture is broken into a series of very narrow strips by image dissection techniques. A number of small identical lenses are mounted side by side, with a line joining their centers slightly inclined from the horizontal as in the lower part of diagram. Each lens forms a separate image of a distant event, and these images appear as a flight of steps shown in upper part of diagram. When received on a metal plate containing a truly horizontal slit, the slit picks up top of picture from first image, a strip next below from second image, etc. Since all images are alike, the various parts of the slit from left to right carry the equivalent of one complete picture. This camera uses 15 lenses to dissect the image.

The single long and narrow strip made of dissected elements of original picture is imaged on negative film as a line so narrow that it is less than the resolving power of the film itself. The film is driven past this line image, and in a direction perpendicular to the line, at a speed of 400 fps. This is done by placing one turn of film on inside of a shallow rotating drum where it is held in position by centrifugal force 16,000 times gravity.

The narrow strip image is drawn into a streak, by the moving film, and variations in photographic negative density at any place across this streak contain all the elements of a complete rectangular picture for the instant of time represented by position along the streak. Negative films need move by only the width of the narrow slit image to produce a complete new frame of the original rectangular picture. Since the camera is shutterless, the motion of the film produces a blurring of the image, but this blurring is limited to the width of the slit.

After processing the negative must be reconstructed into the original rectangular picture by projection back through an optical system similar to that which formed it. This is done by an automatic projection printer that produces an ordinary 16 mm picture print that can be run through an ordinary projector. Negative is enlarged ten-fold in printing to 16 mm size. Film grain imposes a limitation on picture quality but pictures are adequate for scientific study.

The portable camera weighs 60

pounds and can be used with non-luminous or self-luminous objects. It will be used in industrial research and in study of high explosives, electrical discharges, shock fronts and extremely rapid motion in jets and projectiles.

Television Programs Recorded By Motion Picture Camera

A 16 MM MOTION PICTURE CAMERA for recording television programs on film has been developed by the Eastman Kodak Company. The new camera produces movies directly from the face of the monitoring "picture tube" in television broadcasting.

The camera takes pictures at the rate of 24 frames a second. Sound is recorded separately.

Main uses of the new camera in television broadcasting will be: To enable the recorded programs to be reused by the sponsor for institutional public relations and advertising; to record transmitted programs that go out on the air. This might be important for legal purposes.

Another possible major use, still in the experimental stage, is in a television "film network."

Basic camera design features of the camera include: A 1,200-foot film magazine that permits continuous recording of a half-hour program; separate, synchronous motor drives for the shutter and film-moving mechanisms; a coated f 1.6 lens of 2 in. focal length.

The double-chamber magazine is a unit housing both the unexposed and the exposed film. It may be readily removed from the camera.

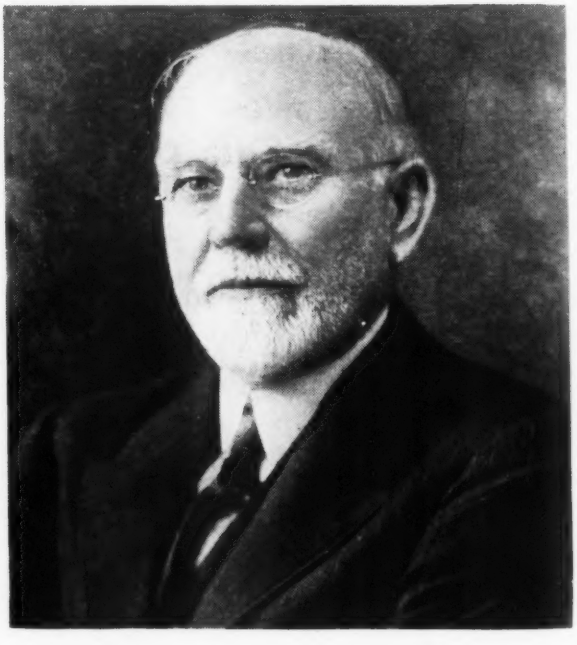
Atomic Physics Division at Bureau of Standards

A DIVISION OF ATOMIC PHYSICS has been formed at the National Bureau of Standards to group activities relating to atomic and molecular physics.

Functions of the new division are promotion of fundamental fact finding research and precise determination of important fundamental standards in the fields of atomic physics. This research will be applied to establishment of primary and secondary standards of reference; for example: Calibrated sources of radiation, meters for dosage, and calibration of instruments used in atomic physics studies. The division is also responsible for development of protective and safety codes for reduction of industrial hazards arising from new applications of physics, and for study of materials and equipment.

Six sections make up the new division: Spectroscopy, Electronics, Mass Spectrometry, Radioactivity, X-Rays and Atomic Physics. The sections are pursuing individual research projects under direction of Dr. E. U. Condon, Bureau's director, who has assumed charge of the new division.

James H. McGraw, Sr.



FOUNDER of the McGraw-Hill Publishing Company, James H. McGraw, Sr. passed away on February 21, 1948, in San Francisco, Calif., at the age of 87.

One of the leading figures in the publishing field and dean of industrial publishers, Mr. McGraw based the expansion of his publishing activities on the five major engineering professions and industries allied with them.

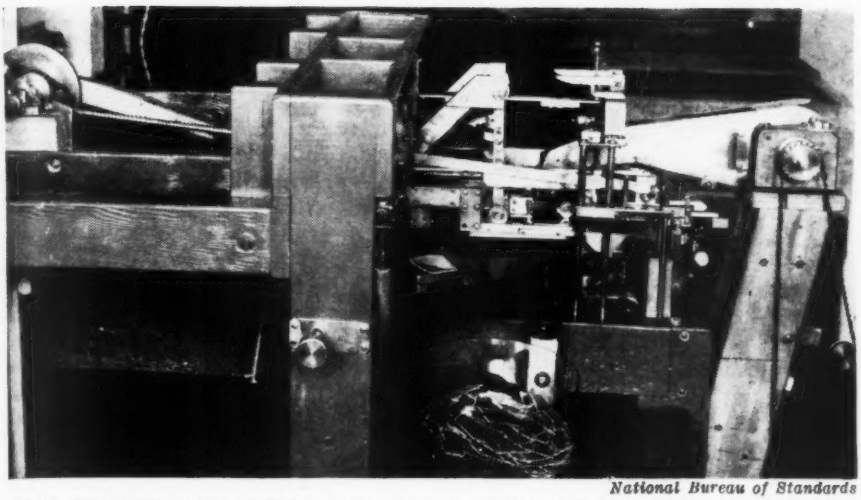
Mr. McGraw was born on December 17, 1860, in Panama, Chautauqua County, New York. He attended the State Normal School at Fredonia, and at the same time, taught at a district school. When Chautauqua County friends started *"The American Journal of Railway Appliances"* and purchased a little paper called *"Steam"*, the temptation was too much. He made the decision which was to change the entire course of his life. But at the height of his career he said, "I believe I would have loved teaching better than publishing."

At any rate, he left his country school house to make his fortune in New York by selling subscriptions. By the summer of 1886, the company owed him $1500, with no prospects for payment. Borrowing $1000 from James Knapp, a wealthy Quaker farmer of Chautauqua County, who had known him when he was a boy, Mr. McGraw took stock for that amount as well as for the $1500 due him, and thereby became a vice-president of the little publishing house. That marked a turning point, for two years later the company paid a 10 percent dividend.

By 1901, he had acquired *"Electrical World"* and *"Electrical Engineer,"* later consolidated as *"Electrical World."* He later added *"Engineering Record," "Chemical and Metallurgical Engineering,"* and *"Electrical Merchandising."*

Upon the death in 1916 of John A. Hill, who had left an engine throttle in the Far West to come East and work on a locomotive paper, the McGraw Publishing Company acquired the Hill publications, thereby adding *"American Machinist," "Power," "Coal Age," "Engineering News,"* and *"Engineering and Mining Journal"* to the group. *"Engineering Record"* and *"Engineering News"* were combined, and the McGraw-Hill organization began with nine national publications.

Mr. McGraw resigned on Dec. 27, 1935, as chairman of the Board of Directors, when his responsibilities were taken over by his son, James H. McGraw, Jr., who assumed the presidency and chairmanship of the Board.



National Bureau of Standards

MICA SPLITTING MACHINE developed at the National Bureau of Standards showing a mica block mounted on the chuck ready for shifting under the accurate splitter.

Mechanical Mica Splitter Developed at Bureau of Standards

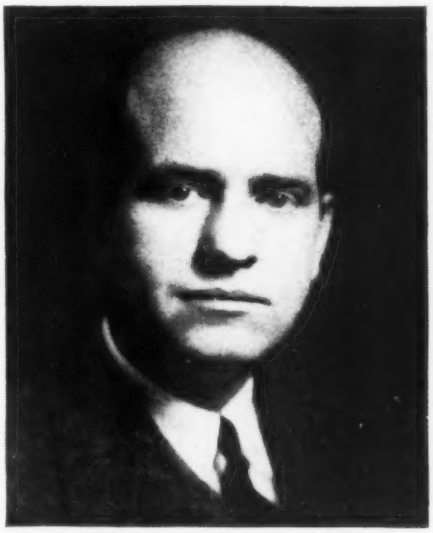
MECHANICAL SPLITTING OF MICA, a mineral which has excellent insulating and heat-resisting qualities and is used to separate condenser plates, insulate electrode elements of vacuum tubes and for many other electrical insulating purposes, has been developed at the National Bureau of Standards. It is a much more feasible means of producing thin films than the tedious hand method commonly employed. The mica splitting machine developed by D. W. Kessler and R. E. Anderson, of the Bureau of Standards, promises to speed up the process considerably and will greatly reduce the period required to train skilled splitters.

Because of common mica imperfections such as waviness or unequal spacing of cleavage planes, no precise control of film thickness has been obtained. With a gage setting for 2 mil films a large percentage of the splits will be near the desired thickness, but occasional films may vary as much as 50 percent. The quality of the machine-split films compares favorably with that of hand-split film.

The machine starts thin sheets by a sharp point entering one edge of the mica block. The operator places a block on the chuck and turns it under the splitter. The master valve at the center of the turntable connects the chuck to the vacuum line to hold the block in position while it is being split.

On traveling forward, a carriage moves the chuck a short distance, causing the splitter gage to raise a film of mica. As the carriage returns to its starting position, two stripper blades pass under the raised film to separate it from the block. At this stage the free end of the conveyor moves down near the loosened film and draws it to the belt which carries it to a receptacle.

When one chuck is shifted into the splitting position, another comes before the operator which, when loaded, is ready for shifting. Since the turntable locks each time a chuck is in the splitting position, the operator must unlock the latch on his left, hoist the carriage tracks with a foot pedal, and shift the newly loaded chuck until it locks under the splitter.



CHEMICAL SOCIETY PRESIDENT—Dr. Charles A. Thomas, executive vice-president and technical director of the Monsanto Chemical Co., St. Louis, Mo., took office as president of the American Chemical Society in January. Dr. Thomas, Project Director of the Clinton Laboratories, Oak Ridge, Tennessee, and active in catapult rocket development, will receive the 1948 gold medal for achievement from the American Institute of Chemists this year.

Metal Films Applied Continuously By Evaporation

CONTINUOUS COATING OF SHEET MATERIALS with metallic films by an evaporation process has been recently developed by National Research Corporation, Cambridge, Mass. While such metals as aluminum have long been applied to the surface of glass or other material by batch methods, their quick, continuous application to continuous rolls at high speeds has not been practical until now. Aluminum, silver, gold, zinc, copper and other metals can be applied to papers, textiles, and various plastic sheets at linear speeds of hundreds of feet per minute.

Because of the extreme thinness of the film, the flexibility of the base material is not affected and the moisture vapor transmission is reduced. Adhesion varies with the type of material. Finish is limited only by that of the base and on polished surface brilliance exceeds that of foil.

The process is now in a stage of advance pilot plant operation and is estimated to be more economical than rolling foil for similar applications.

At present, the process is used in decorative materials of great variety—wrapping papers, ribbons and novelty fabrics. It is believed that functional packaging applications can be developed. Various electronics uses have also been tested. Capacitors or condensers formed from zinc-coated paper have self-healing qualities that promise long life to such units.

Parachute Enables High-Speed Bail-Out

PILOT SAFETY DURING HIGH-SPEED JUMPS has been increased by a new parachute designed at the Navy Parachute Experimental Unit, Naval Air Station, El Centro, Calif. Research in problems of escape from fast moving aircraft is directed by the Airborne Equipment Division of the Bureau of Aeronautics.

This parachute has a standard canopy with a 6 ft dia moveable "cap" in the center. Under high stresses, the cap extends on elastic cords and spills air. Securing loops prevent cords from stretching more than 18 inches. As the chute slows, the cap gradually closes and rate of descent is decreased.

During tests, the parachute is dropped in a dummy bomb. After falling a short distance, parachute unfurls and bomb becomes test weight. Tests have been completed at speeds

to 473 mph, compared with a maximum speed of 239 mph for present personnel parachutes.

High-speed parachutes will be necessary even though fighter planes have been designed with pilot ejection seats or ejectable capsules (see PRODUCT ENGINEERING, Jan 1948, page 150). Parachutes designed by two private contractors have been tested up to 385 mph; one contains mesh sections and the other is a variation of German "ribbon" chute.

Photo-Memory Unit Developed For Navy Electronic Computer

A "PHOTOGRAPHIC MEMORY" UNIT developed for use by the Navy in a large electronic calculator has been announced by the Eastman Kodak Company. The unit when completed will be able to "read" and "memorize" mathematical data at a speed of one thousand numbers of 12 digits each every second.

The device Kodak is building in Rochester will be part of an electronic computer being developed by the Massachusetts Institute of Technology at Cambridge, Mass. Supervision of the calculator and memory projects is under the Special Devices Center of the Office of Naval Research, Sands Point, Port Washington, L. I.

Key to the memory is the use of spots. They are clear and opaque rectangles, about 0.01 in. square, recorded on strips of film. These spots representing digits will come either from the calculating machine or from a hand-operated keyboard. Then these numbers, such as the results of complicated mathematical computations, will be recorded ("memorized") on the film.

The electronic calculator will control and present information in the form of spots of light on the face of a cathode-ray tube. This tube resembles the viewing tube of a television receiver. A special camera, meanwhile, will photograph each pattern of spots as it is produced by the moving electronic beam within the tube. Thus the numbers are permanently recorded.

Conversely, after the film has been speedily developed, the unit will be able to supply ("read") the information back into the calculator for further computations. The unit will move the developed film through an optical system. This will project the recorded data onto photo-electric tubes. These tubes, in turn, will supply electrical signals—corresponding to the numbers on the film—to the calculator.

For auxiliary use with the unit, the company will build an apparatus to convert automatically the information recorded on film into tables of numbers and graphs suitable for direct studying.

Industrial Applications Feature Midwest Plastics Meetings

AUTOMOTIVE, AIRCRAFT and many other industrial applications were described at two important plastics conferences held recently in Chicago and Detroit. Indicating use of new and accepted plastic materials by design engineers, applications were exhibited at the low pressure industries meeting, Society of the Plastics Industry, at the Edgewater Beach Hotel, Chicago; and at the annual conference of the Society of Plastics Engineers, held at the Rackham Memorial, Detroit.

REINFORCED PLASTICS

Reflecting the changing attitudes of of the "low pressure" division of the plastics industry, a proposal to change the name of this group to the Reinforced Plastics Division was put before the meeting. Several reasons were advanced for this suggestion. One, was the evident shift of products from the decorative and novelty class to accepted industrial components. The second, and possibly more important reason, was the use of "high pressure" methods in this industry. The interior panels of the DeSoto suburban station wagon was produced by the Briggs Mfg. Co. (see "Recent Developments in Plastics and Rubber", PRODUCT ENGINEERING, April 1947) by a combination of low pressures and high pressure molds.

In the exhibit held concurrently with the technical meeting, a wide variety of industrial components were shown. Electrical aircraft parts, and aircraft forming tools, molded of glass cloth and mat with polyester resins were shown by Lawrence Wittman of the Republic Aviation Corporation. Laminated Plastics Inc., Cleveland, exhibited a series of bus and train coach seat parts, also molded of glass cloth and special "low pressure resins". The same company showed a series of electrical parts—a motor armature coil support, dc motor computator cap, an armature end punching, and a dc field coil bobbin.

Of special interest to both the plastics and the automobile industry, was the sun visor for the new Studebaker, made of glass fiber mat and polyester resin. This was developed by Dr. Ray Nelson, Naugautauck Division, U. S. Rubber Company. Molded in metal dies, and using contact pressure only, the piece showed unusual rigidity. The extremely glossy finish and light weight of the part also offered advantages to automobile manufacturers. In the same field, automotive fender skirts, of glass fiber and polyester resin, were molded by the Bolen Mfg. Co., San Francisco.

Reinforced plastics were also used for many aircraft parts. Shown by Zenith Plastics, Los Angeles, were a window frame for a new Douglas airliner, an air duct, an antenna mast, and stabilizer fin tips for a new Lockheed airplane. The Glenn L. Martin Co. exhibited a wide variety of interior aircraft parts made of honeycomb cores faced with plywood. Of particular interest was a door section made for the Grummann Mallard.

Technical progress in the low pressure industry was described in several papers. Dr. F. L. Minnear, Shellmar Products Corp., spoke on "Evaluation of Thermoset Laminated Sheet Products." This talk did much to reinforce the idea of tying together the low and high pressure laminating industries. After describing the early phenolic laminates, Dr. Minnear discussed the polyester, styrene copolymers and melamine laminates. Phenolic resins are predominantly used in the high strength, high pressure laminates. However, they are limited by their dark color. Melamine resins, although relatively high in cost, give a wide range of decorative shades. Polyester and styrene copolymer resins are most used in the low pressure and contact pressure methods, because of their inherent chemical ability to set up. Mr. W. Burdette Wilkins, a consultant of Ridgewood, N. J., discussed the economics of reinforced plastics. Mr. Wilkins listed the general types of articles which lend themselves to low pressure molding: Large articles to be produced in small quantities; parts with compound curves, or parts requiring high strength, low weight, di-

electric strength or weather resistance superior to metals, or high-production plastics.

Plastics Engineers

The annual meeting of the Society of Plastics Engineers gave abundant evidence that engineering and research techniques in this industry are rapidly maturing. Engineering applications in the automotive industry were described by W. M. Phillips of the General Motors Corporation. Mr. Phillips said that about a hundred plastics parts weighing about five pounds were used in the average automobile. The trend is continuing, and Mr. Phillips noted that vinyls and polystyrene are improving their position in this respect. Also promising is the use of plastic "envelopes" in stamping operations. In addition, certain laminates are being introduced, and show promise of replacing several steel parts. Nylons and other plastics show increasing promise for use in upholstery.

An original research study, showing a mathematical approach to the subject, was presented by Professor A. H. Dietz of the Massachusetts Institute of Technology, in his talk on stress-time relations in plastics. Professor Dietz reviewed the classical theories and mechanical analogs of the behavior of visco-elastic materials as a function of time. To apply these theories to plastics, testing equipment was developed to provide control over rates of crosshead, load application and rates of either ordinary strain or true strain. Professor Dietz described the servo controls, special fixtures such as self-aligning locking jaws for reversed tension-compression cycles, and methods of using extensometers to control the motion of the machine. Using this equipment and plastics of various types, from hard and rigid to soft and extensible, typical stress-time relationships were presented.

A physical and mathematical analysis of plastic flow was presented by James Bailey of the Plax Corporation in a talk on stretch orientation of polystyrene. Mr. Bailey pointed out that stretching produces the greatest strength in the direction of stretch, and is useful only in monofilaments. Large sheets require bilateral strength. Compression molding can produce sheets that are tough in two directions —therefore weak in one direction. Longitudinal stretch produces a reduction in the coefficient of expansion and great optical virefringence. Polarascopic studies of strains showed that hot stretching can produce ten times as much virefringence as rupturing stress at room temperatures. Equal bilateral stretching produces no strain.

A new research method for studying the degradation of plastics was described by F. W. Reinhart, National Bureau of Standards. The object of the method is to determine the specific chemical reaction involved in the degradation of the plastic and how these reactions are affected by the intensity of environmental conditions. This method assumes that physical changes, observed visually and in tests, result from changes in the chemical structure of the resin and from loss and changes in the compounding ingredients. If true, application of this method should be an absolute check on the resistance of any given plastic material to moisture, heat, and other environmental conditions.

Other talks of interest to design engineers included: "Styrene Polymers and Copolymers for Industry" by R. Fordyce, Monsanto Chemical Co.; "Industrial Uses for Nylon", by J. E. Teagarden, E. I. duPont de Nemours & Co.; "The Theory of Colorimetry and its Application to Color Standardization", by Miss E. Dolton, Interchemical Corporation; "Phenolic Resins in Synthetic and Natural Rubber", by Jay Searer, Durez Plastics and Chemical Co.; and "Glass Bonded Mica in Molded Form" by Dr. F. L. Yerzley, Mycalex Corporation of America.

HEADS PLASTICS ENGINEERS—J. H. Du Bois, executive engineer of the Shaw Insulator Company, Irvington, N. J., was elected president of the Society of Plastics Engineers, at its annual meeting in Detroit. Mr Du Bois is an active member of the plastics industry, having been formerly associated with the plastics division of the General Electric Company. He is the author of two books in this field: "Plastics", and "Plastics Mold Engineering".

MEETINGS

March 1-4
American Society of Mechanical Engineers—Spring Meeting. St. Charles Hotel, New Orleans, La.

March 3-5
Society of Automotive Engineers—Passenger Car and Production Division Meeting, Hotel Book Cadillac, Detroit, Mich.

March 4-6
Optical Society of America—Annual Winter Meeting, Hotel Pennsylvania, New York, N. Y.

March 22-24
Chicago Technical Societies Council—Chicago Production Show, Hotel Stevens, Chicago, Ill.

March 30-31
Society of Automotive Engineers—Transportation Division Meeting. Hotel Bellevue-Stratford, Philadelphia, Pa.

April 5-8
National Association of Corrosion Engineers—Annual Conference and Exhibition, Jefferson Hotel, St. Louis, Mo.

April 5-8
Southern Machinery and Metals Exposition—Annual Meeting and Exhibition, Atlanta Municipal Auditorium, Atlanta, Ga.

April 13-15
Society of Automotive Engineers—Aeronautic and Air Transport Division Meeting, Hotel New Yorker, New York, N. Y.

April 15-16
Metal Powder Association—Annual Spring Meeting, Drake Hotel, Chicago, Ill.

April 26-30
American Management Association—Annual Conference and Packaging Exposition, Public Auditorium, Cleveland, Ohio.

May 3-5
Industrial Furnace Mfg. Association—Annual Meeting, The Homestead, Hot Springs, Va.

May 3-7
American Foundrymen's Association—Annual Meeting, Philadelphia, Pa.

May 3-14
British Industries Fair—Sponsored by Export Promotion Dept., held in London and Birmingham.

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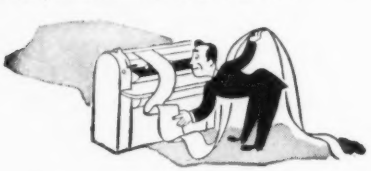

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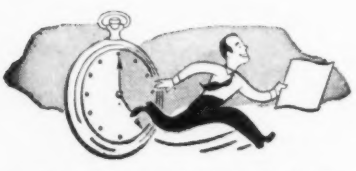

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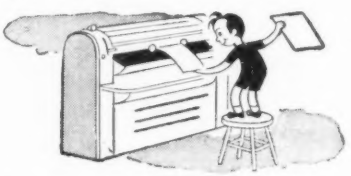

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Charts for Designing Long-Short Addendum Spur Gears

WAYNE H. BOOKMILLER
Gear Engineering Division, General Electric Company

ENLARGEMENT OF PINIONS to avoid undercutting the teeth or to increase their strength, or both, has become standard practice.

The gear that mates with an enlarged pinion must be mounted at a greater center distance or must be reduced in diameter. The latter method is usually preferred.

Gear sets made with long addendum (enlarged) pinions and short addendum gears can be designed so the beam strengths of gear and pinion are either equal or related by a fixed ratio. The family of curves in Fig. 1 shows the addenda necessary to have the same form factor Y for the pinion and gear. The actual addenda are found by dividing the curve values by the diametral pitch.

Since the Y factor is based on the worst application of load, and a stress concentration factor K is included, the Lewis Equation as now used is; where

S = Root stress, psi
W_T = Tangential driving pressure, lb
P_D = Diametral pitch
F = Face width, in.
K = Stress concentration factor
Y = Form factor of tooth

$$S = \frac{W_T \times P_D \times K}{F \times Y} \qquad (1)$$

The form factor Y and the stress concentration factor K, for use in Eq. (1), are found on Figs. 2 and 3 respectively. These curves apply to long-short addendum gear sets designed for equal Y factor pinions and gears.

As an example of the use of Eq. (1) and Figs. 1, 2 and

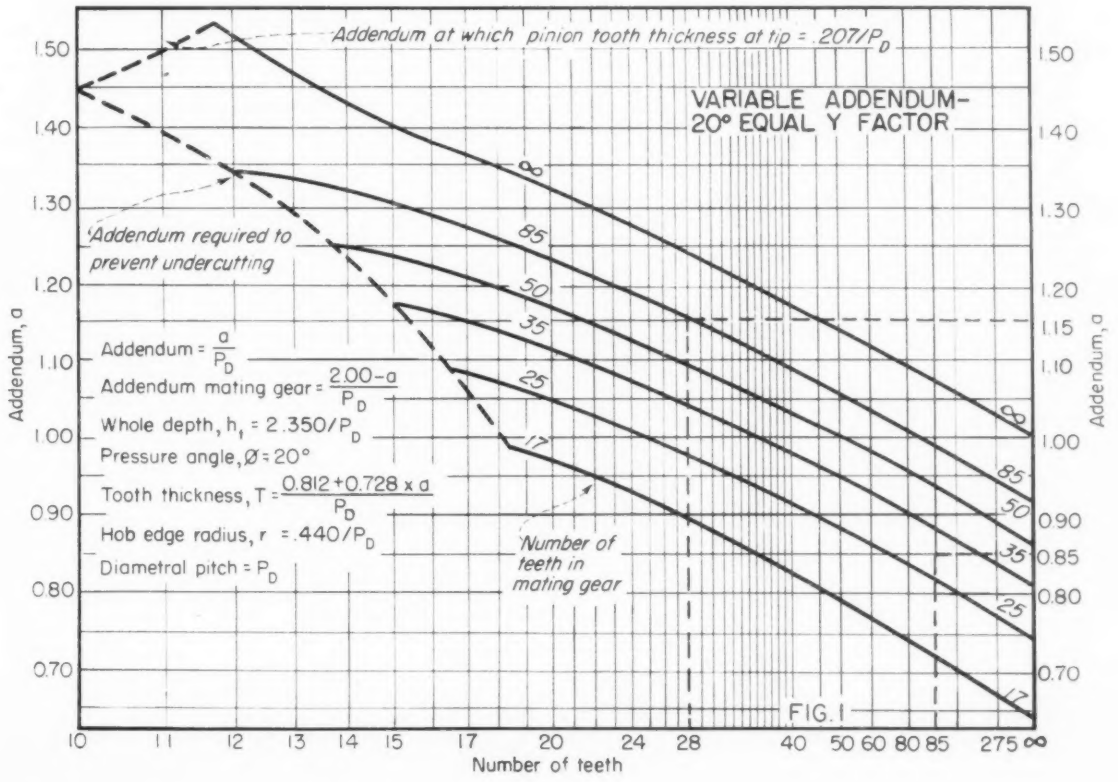

FIG. 1

Die Casting by

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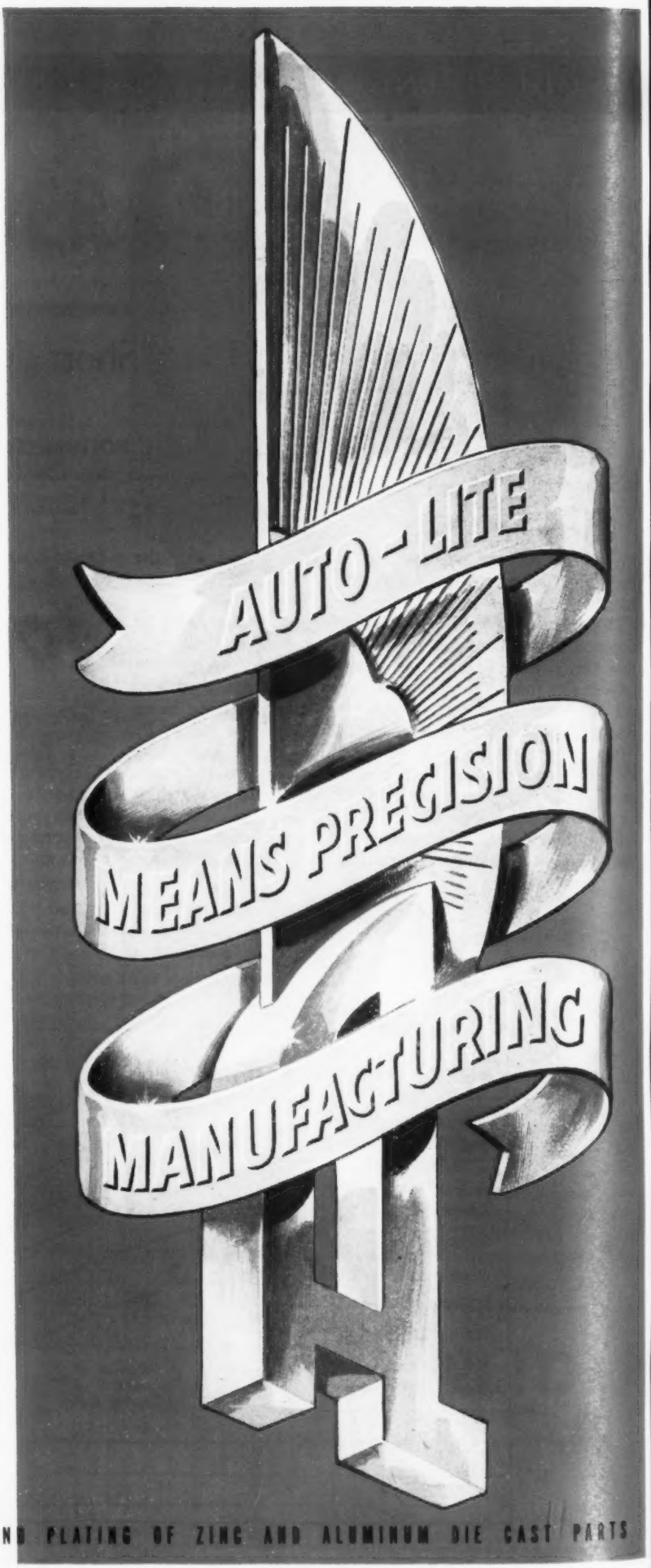

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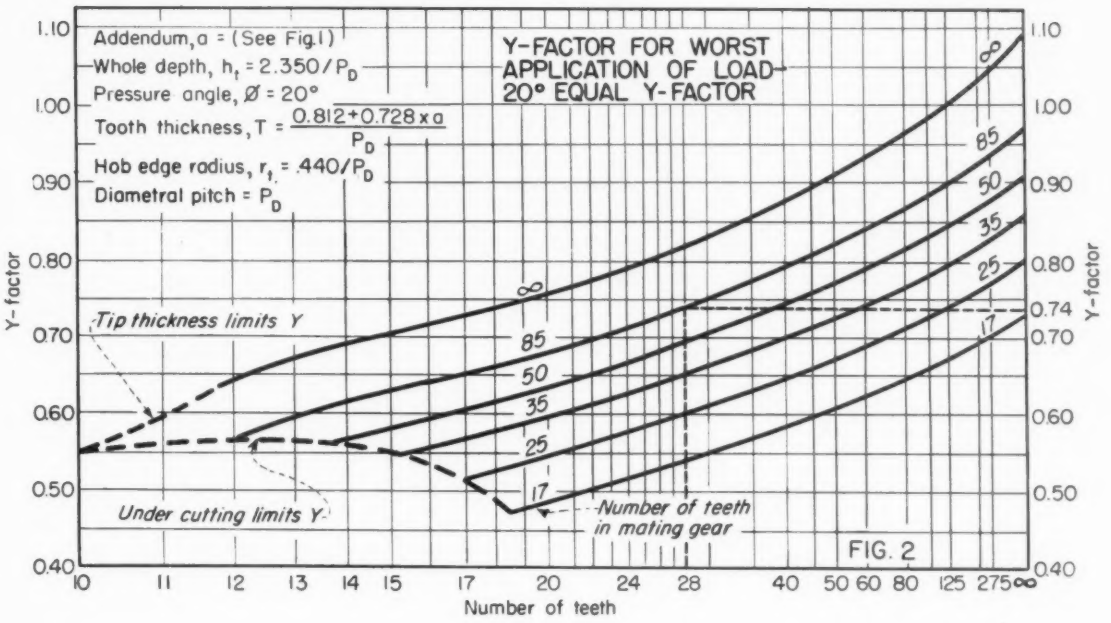
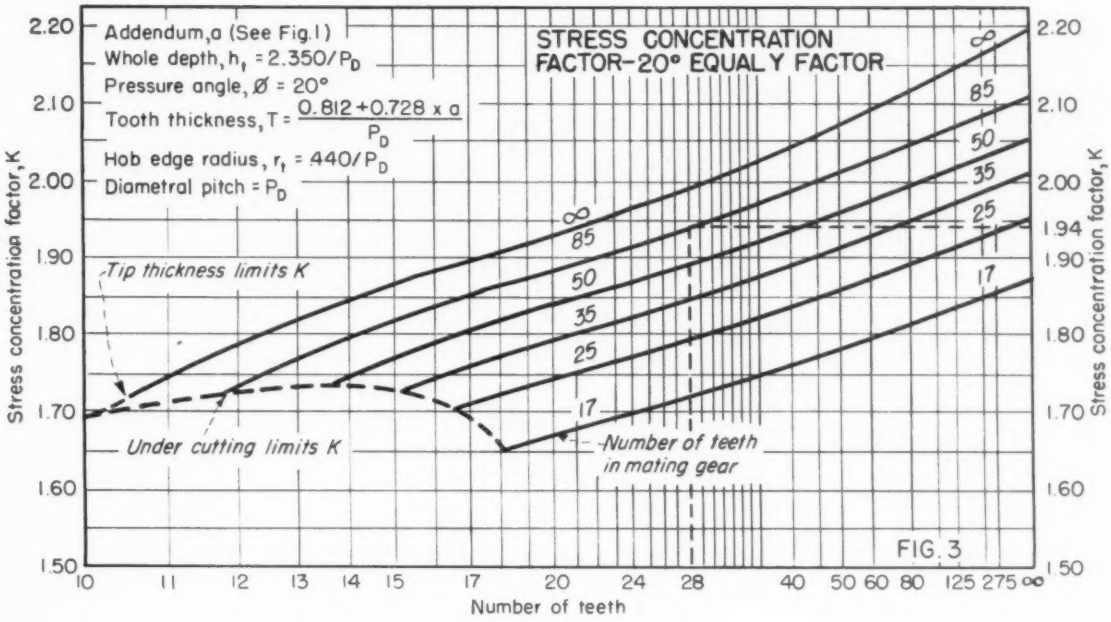
CASTING, MACHINING, FINISHING AND PLATING OF ZINC AND ALUMINUM DIE CAST PARTS


Addendum, a = (See Fig.1)
Whole depth, $h_t = 2.350/P_D$
Pressure angle, Ø = 20°
Tooth thickness, $T = \frac{0.812+0.728 \times a}{P_D}$
Hob edge radius, $r_f = .440/P_D$
Diametral pitch = P_D
Y-FACTOR FOR WORST APPLICATION OF LOAD-20° EQUAL Y-FACTOR
Tip thickness limits Y
Under cutting limits Y
Number of teeth in mating gear
Y-factor
Number of teeth
FIG. 2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Tooth thickness, $T = \frac{0.812+0.728 \times a}{P_D}$
Hob edge radius, $r_f = .440/P_D$
Diametral pitch = P_D
STRESS CONCENTRATION FACTOR-20° EQUAL Y FACTOR
Tip thickness limits K
Under cutting limits K
Number of teeth in mating gear
Stress concentration factor, K
Number of teeth
FIG. 3

3, consider the following gear set:

	Pinion	Gear
Number of teeth	28	85
Diametral pitch,	20	20
Pressure Angle, deg	20	20
Pitch diameter, in.	1.400	4.250
Speed, rpm	20,000	6,600
Face width, in.	½	½
Horsepower	15	.....

$$\text{Pinion addendum} = \frac{a}{P_D} = \frac{1.150}{20} = 0.0575 \text{ in.}$$

$$\text{Pinion tooth thickness} = \frac{0.812 + 0.728 \times 1.15}{20} = 0.0825 \text{ in.}$$

$$\text{Gear addendum} = \frac{a}{P_D} = \frac{0.850}{20} = 0.0425 \text{ in.}$$

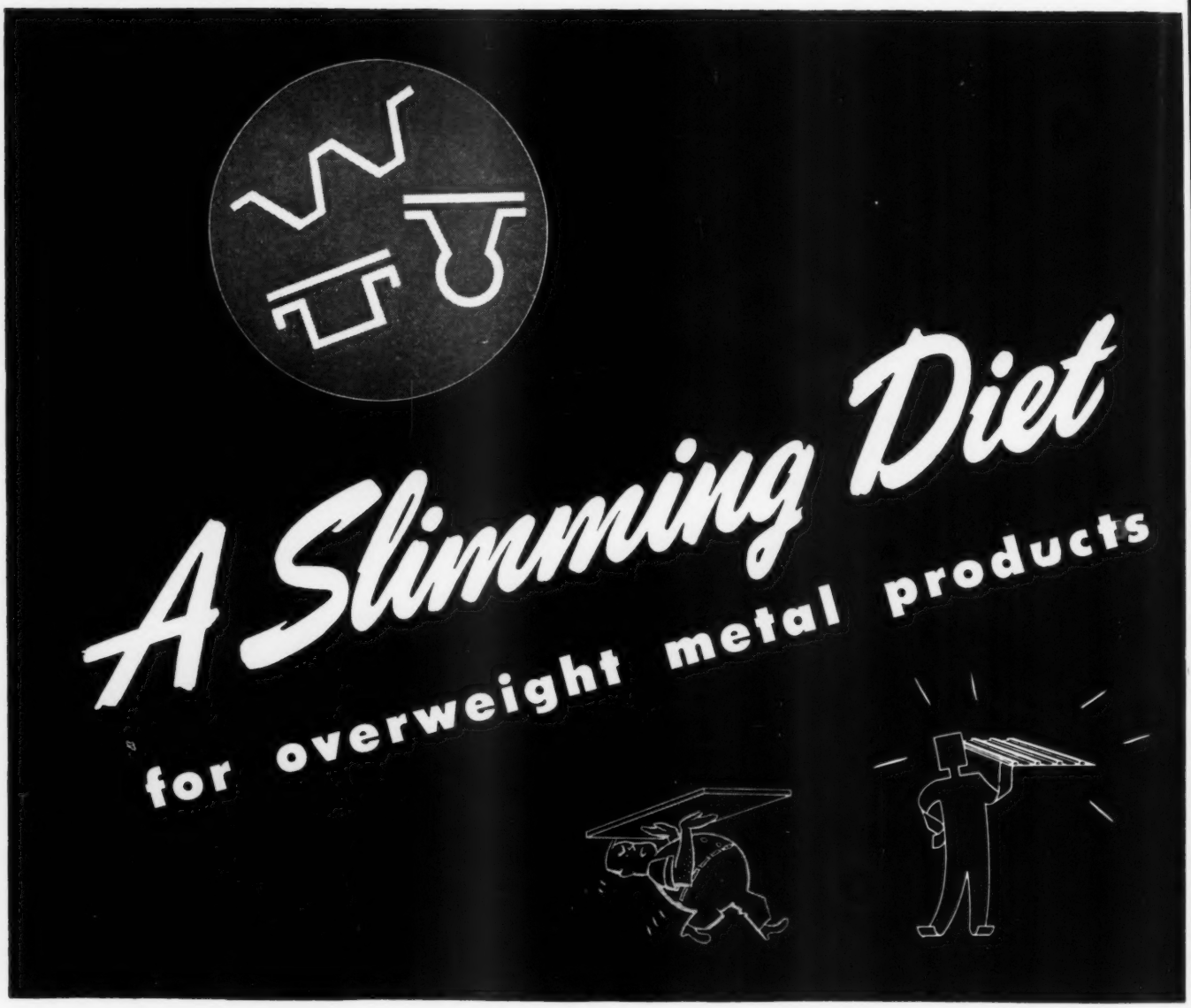
$$\text{Gear tooth thickness} = \frac{0.812 + 0.728 \times 0.850}{20} = 0.0715 \text{ in.}$$

$$W_T = \frac{15 \times 33{,}000 \times 12}{20{,}000 \times 2\pi \times 0.7} = 67.5 \text{ lb}$$

$Y = 0.740$ (Fig. 2) $\qquad K = 1.940$ (Fig. 3)

$$S = \frac{W_T \times P_D \times K}{F \times Y} = \frac{67.5 \times 20 \times 1.94}{0.500 \times 0.740} = 7{,}080 \text{ psi, for pinion}$$

The gear root stress can be calculated in a similar manner and will be approximately equal to the pinion root stress.



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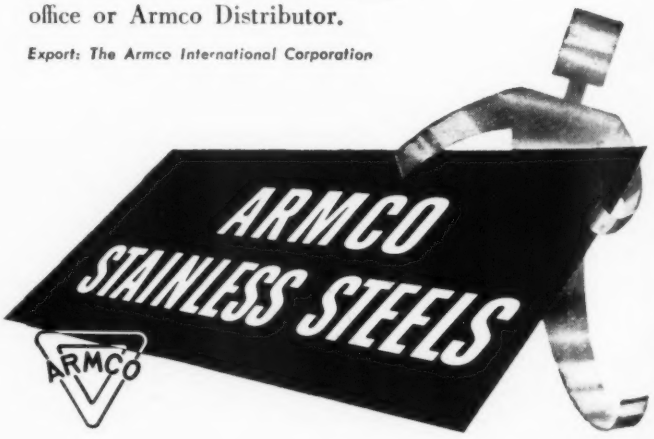

NEW MATERIALS AND PARTS

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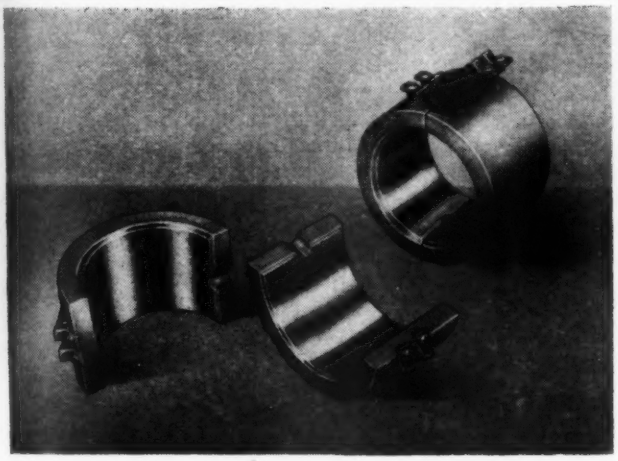


Electric Cylinder Heater

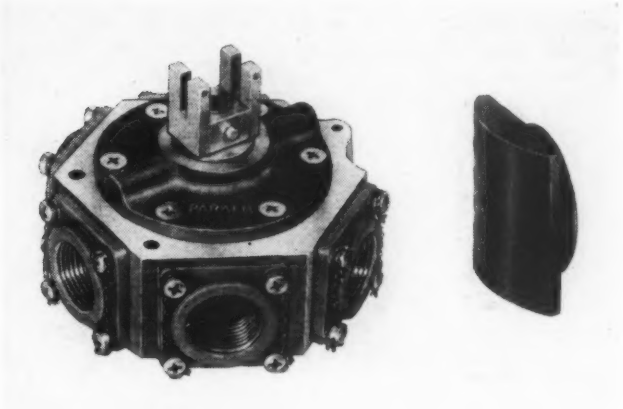
Edwin L. Wiegand Co., 7535 Thomas Blvd., Pittsburgh 8, Pa.

The Chromalox cast-in-aluminum tubular electric heater has been especially designed for clamping to cylindrical surfaces. The unit provides uniform heat that is ideal for application to plastic extrusion and injection molding machines. It is applied to barrel or cylinder as two individual semi-circular units held together by clamping bands. A perfect fit, achieved by machining the inside diameter, minimizes hot spots and assures maximum conduction of heat to cylinder. The aluminum absorbs and distributes heat from the unit, making possible lower operating temperatures and longer heater life. Heater diameters to fit various barrel diameters and temperature requirements are available.

Fluid Pulsation Dampers

Equipoise Controls, Inc., 100 Stevens Ave., Mt. Vernon, N. Y.

Gage fluctuations caused by pressure pulsation and vibration can be smoothed out through the use of snubbers. This device comprises a threaded fitting containing a porous, stainless steel disk $\frac{1}{8}$ in. thick at one end. The disk provides relatively large filter surface that protects the indicator mechanism against shock, clogging and wear and permits accurate indication of changes in pressure. Specially designed disks are available for air, oil, water, mercury and other mediums. Maximum differential pressure rating of the standard types is 500 psi although units can be supplied with ratings up to 5,000 psi differential. The housing has either $\frac{1}{4}$ or $\frac{1}{2}$ in. threads, male at the disk end and female at the other.

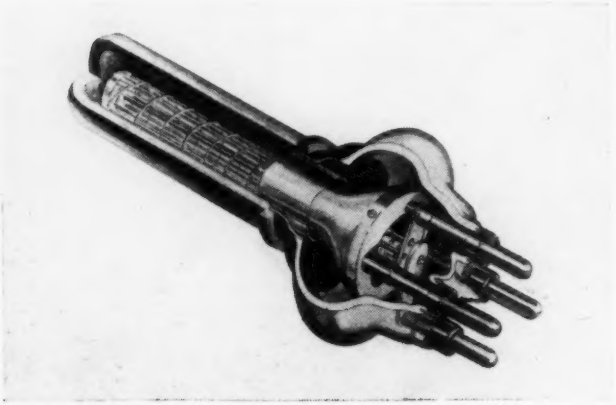


Rotary Selector Valve

The Parker Appliance Co., 17325 Euclid Ave., Cleveland, Ohio

Selector valves of "balanced" design utilize line pressure in sealing various combinations of as many as five separate lines. Graphite sealing inserts are ground to a radius corresponding to the rotor cavity in the valve body. Bonded to the inner side of each insert is a synthetic rubber sac containing a light spring. As the insert is rotated over a port, the spring provides initial contact with the valve body. Additional pressure is supplied by line fluid that enters the sac by a hole drilled through the graphite. This design is suited to all non-shock pressures up to 50 psi. For higher pressures, where the sac might be a limiting factor in corrosiveness and temperature, the spring alone is the sealing medium. Valves

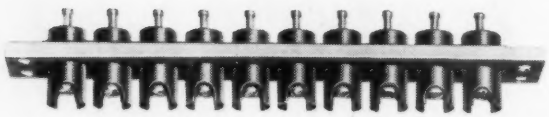
can be furnished for hand or electric motor operation. Adaptable to chemical and process service, the valves will handle air, gases and liquids up to 500 F. Sizes from $\frac{1}{2}$ to $1\frac{1}{4}$ in. are available in aluminum, bronze, stainless or special alloy steels.



Heavy-Duty Electron Tubes

Machlett Laboratories, Inc., Springdale, Conn.

The severe electrical and mechanical conditions encountered in industrial RF heating and communications service can be withstood by a line of five specially designed electron tubes. Design characteristics that distinguish these tubes from existing types are: Heavy wall high conductivity copper anode, one-piece high conductivity copper grid and filament support terminals, filament spring design that minimizes bowing and increases filament life, strong self-supporting grid for uniform electron control, and special design glass contour that provides long leakage path and efficient cooling. Ratings of the tubes vary from 5 to 50 kw. All but one are available in both air and water cooled models.



Combination Terminal Block

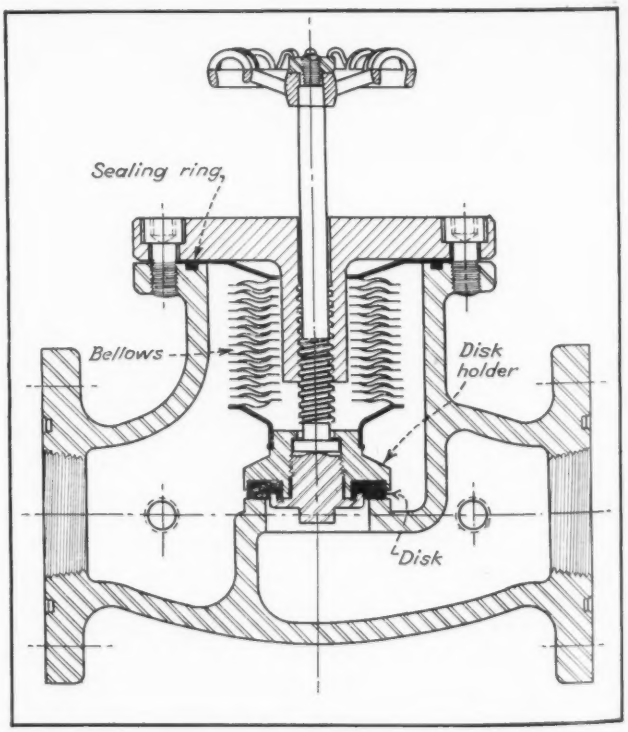
Curtis Development & Mfg. Co., 1 N. Crawford Ave., Chicago 24, Ill.

A feed-through terminal block that combines both screw and solder terminals is now available for sub-panel and chassis construction applications. Each terminal is mounted in bakelite and securely held in a metal strip. The exact number of terminals desired from 1 to 16 can be specified as each block is factory assembled to meet specific requirements. The overall length of a 10-terminal block is $7\frac{1}{4}$ in., with a height of $1\frac{5}{16}$ in., including screw and solder connections and a width of 1 inch. Ample clearance and leakage distances are provided for use in circuits carrying up to 300 v, 20 amperes.

Ellipse Drawing Stencils

Rapidesign, Inc., Glendale, Calif.

A minimum of time is required in making ellipses with the four plastic template sheets contained in Ellipse Set Folio No. 401. Ellipse sizes are $3\frac{1}{8}$ to $2\frac{3}{4}$ in. major axes with a projection range of 20, 30, 45 and 60 degrees. Sizes and center lines are clearly printed on the negative side to prevent wearing off and to simplify selection of the proper ellipse. All edges on the $4\frac{3}{4} \times 9\frac{3}{4}$ in. sheets are cut smooth to insure a neat, clean drawing. The reinforced folio keeps instruments clean and handy.



Vacuum Tight Valve

Kinney Mfg. Co., Boston, Mass.

Model 30 vacuum valve has been developed to overcome the corrosive action of vapors produced in mercury boilers, mercury arc rectifiers and many high vacuum

newest!

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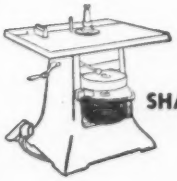
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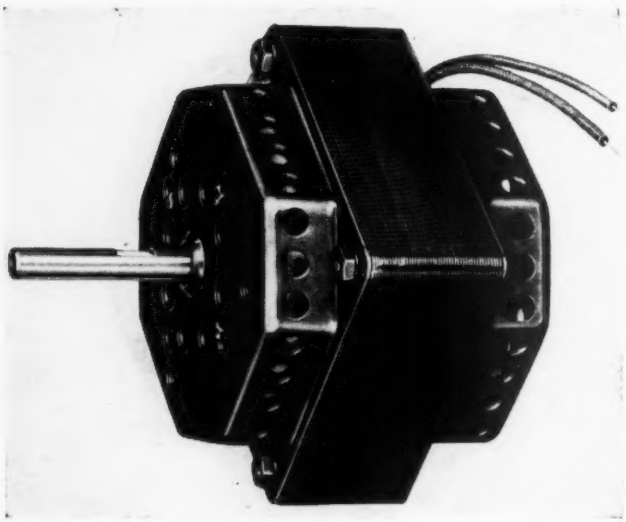
systems using mercury diffusion pumps. The body is cast medium carbon steel with a high test iron cover. Bellows and disk holder are stainless steel and disk and sealing rings are silicone rubber. Design features include replaceable bellows assembly, large port and flow areas, gage connections on inlet and outlet sides of body and two body styles, one for screw or flange connections, the other for welded pipe connections. Both body styles are obtainable in 1, 1½ and 2 in. sizes; 3 and 4 in. sizes are available for welded connections.



Compact Induction Motor

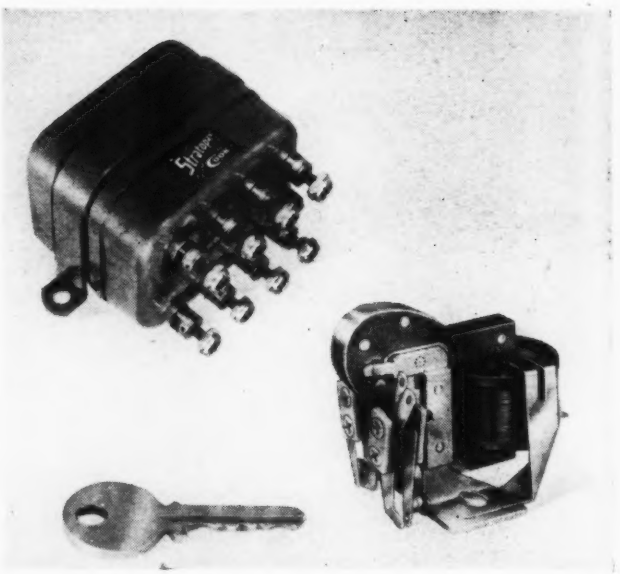
General Industries Co., Elyria, Ohio

The RM-4 Smooth Power shaded-pole induction motor has been redesigned to meet the increased power requirements of wire and tape recording units. Originally used for powering record changer and recording assemblies, the motor now has a bottom cover and locating device that insure rotor alignment. Dynamic balancing of the rotor provides freedom from vibration, noise and magnetic field radiation and special emphasis has been given to cooling and lubrication. Maximum torque is 12 oz-in. when operating at 1720 rpm on 60 cycle a-c current. The unit is 3⅜ in. square with an overall height of 3⅛ in. including bearing bushings. Drive shaft is $\frac{5}{16}$ in. OD and total weight is 4½ pounds.

Photoelastic Resin

Westinghouse Electric Corp., 306 Fourth Ave., Box 1017, Pittsburgh 30, Pa.

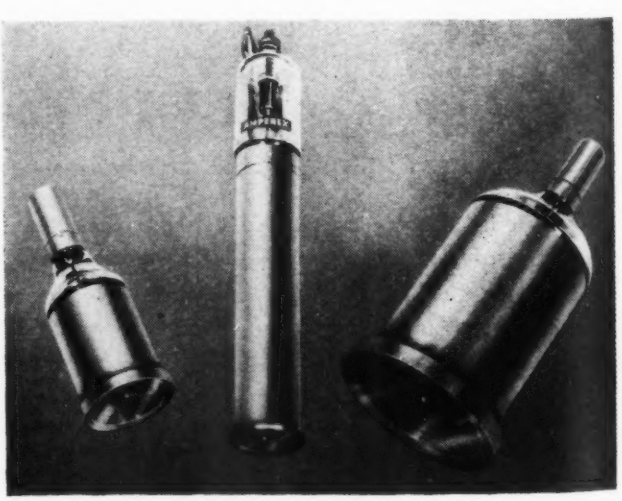
A special process permits large sections of a photoelastic resin of the Fosterite family to be cast for three-dimensional studies of "frozen stresses." One piece measures 6 in. in diameter and 3 ft. long allowing production of sizes and shapes suitable for any kind of three-dimensional study. In the past, plastic models made of ordinary photoelastic resins have been limited to pieces 1⅛ in. thick. The resin is 35 percent more optically sensitive for "frozen stress" measurements than the standard material and does not fade in usefulness with time. Models are made up from the resin, heated, loaded and then cooled gradually to room temperature. When viewed through polarized light, the plastic replica shows vividly the location, direction and magnitude of stresses.



Versatile Precision Relay

Cook Electric Co., Chicago 14, Ill.

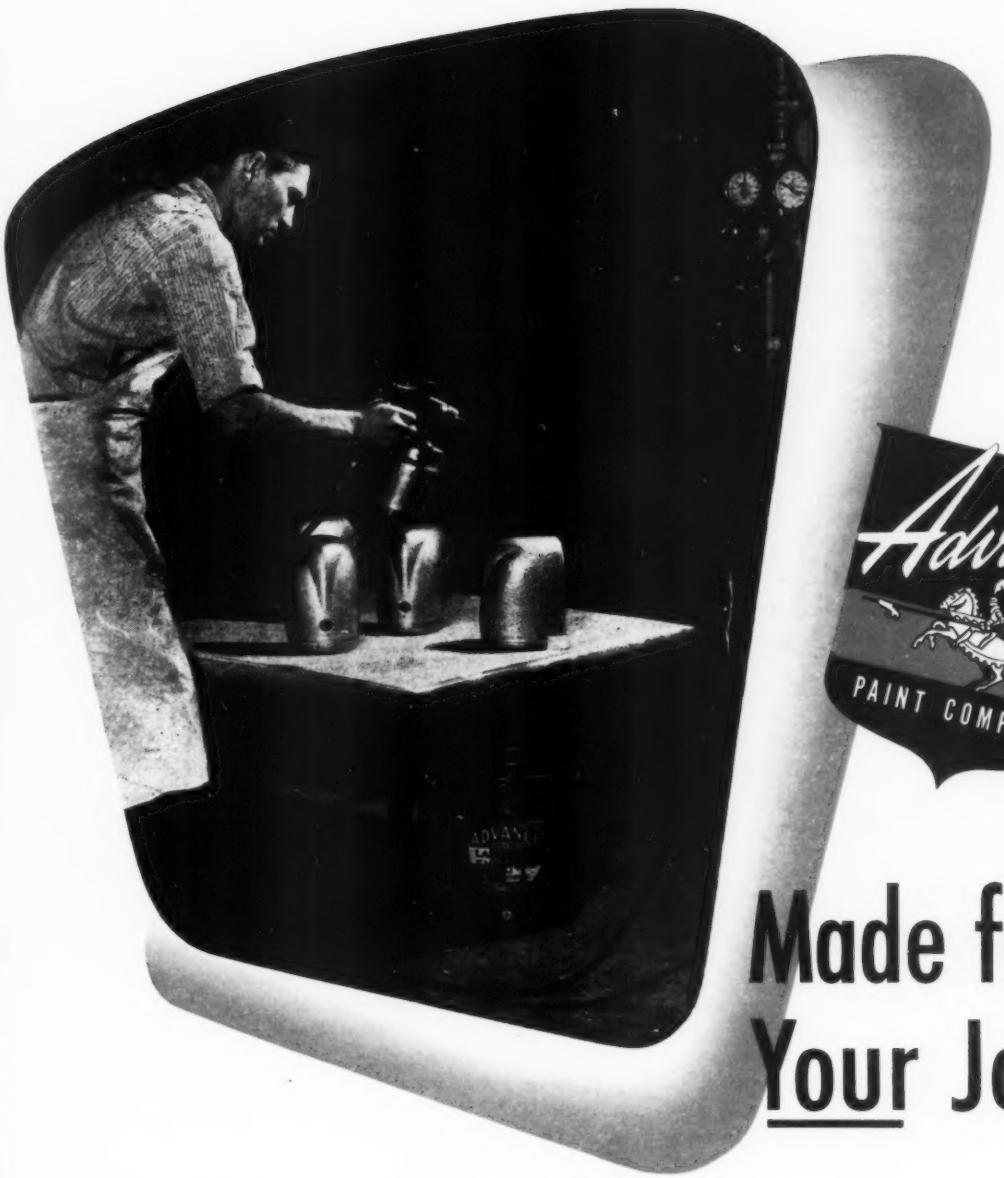
The Hy-G miniature 400 cycle relay has been designed for optimum performance and is adaptable to a variety of contact combinations. The laminated magnetic circuit is so arranged that a very sharp rise in force occurs just before the armature seals in. Maximum stability under vibration and shock conditions is thus assured. Operating voltage range is from 25 to 400 v and the volt-ampere effeciency is high. Coil is impregnated with a silicone compound and heat is conducted through copper bars to radiating fins. Spring assemblies are arranged in two pile-ups, one on each side of the actuating bars, assuring balanced armature pressure. The relay is regularly furnished either with a plain mounting or Stratopaxed. The latter encased unit measures 3 x 2¼ x 2¾ inches.



Radiation Counter Tubes

Amperex Electronic Corp., 25 Washington St., Brooklyn 1, N. Y.

The mica window of radiation counter tubes is being sealed to chrome iron cathodes by a special process that permits uniform quality in volume production. No gaskets or organic waxes are necessary and overall tube



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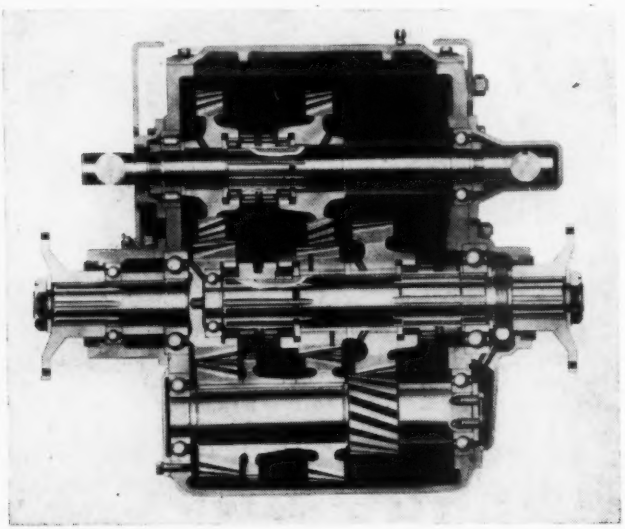
dimensions can be a minimum for any given window size. A special construction permits enclosure of the exhaust tip in a cap of standard dimensions. The beta, X-ray gamma series of tubes, illustrated, embodies these improved techniques. Included in this group are self-quenching counters with operating voltages as low as 250 v and others so completely temperature insensitive that operation in the range from —70 to + 120 C is permissible. Tubes with halogen fillings are also available. The absence of organic quenching vapors gives these tubes an almost unlimited life expectancy.



Auxiliary Transmission

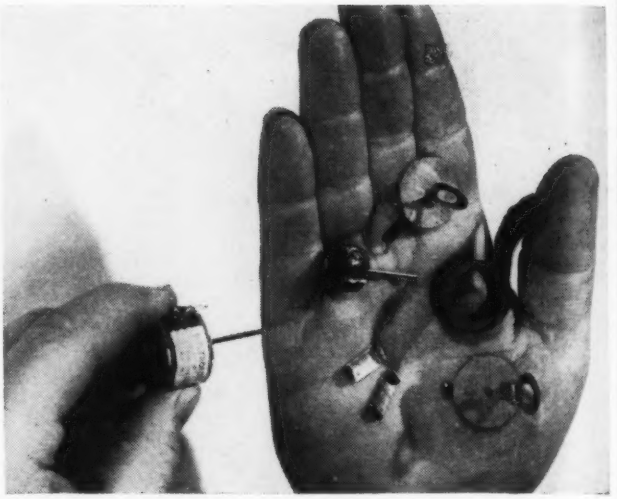
Fuller Mfg. Co., Kalamazoo, Mich.

Model 3-T-92 three-speed auxiliary transmission offers in a single unit, all of the advantages normally provided by auxiliary transmissions with top-mounted power take-off, and at the same time minimizes the inherent weaknesses of this combination. For rigidity and strength the power take-off gears and shaft are built directly into the transmission case where the section of the casting will permit the application of full engine torque. The ratios available form the power take-off vary with the direction of rotation of the shaft. When turning in the same direction as the input shaft of the auxiliary transmission, the ratio is 1.175 to 1. When rotation is in the opposite direction, the ratio is 1.345 to 1. Projections at each end of the power take-off shaft permit coupling of flanges or sprockets bored to $1\frac{3}{4}$ in. ID and keywayed for a $\frac{3}{16}$ x $\frac{3}{8}$ in. straight key.

Balanced-Alloy Tool Steel

Amalgamated Steel Corp., Newburgh Mill District, Cleveland 5, Ohio

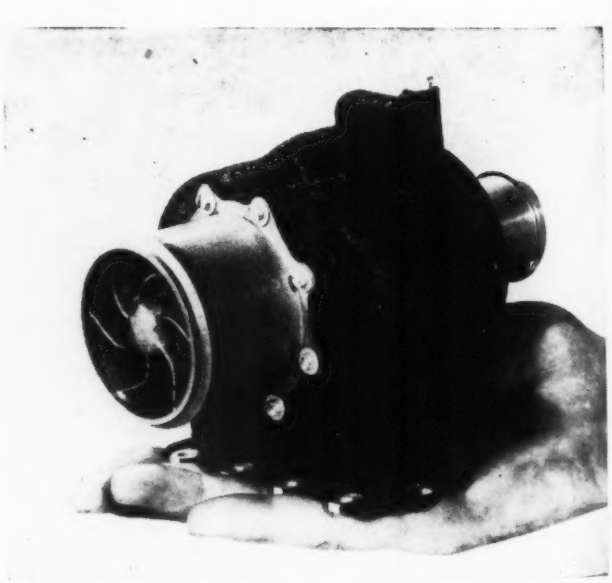
Tool Room Specialty Steel has high resistance to wear and exhibits unusual toughness and strength with ductility to resist fatigue. This general purpose material can be quenched in oil, water or salt bath making it possible to achieve nearly all degrees of hardness. Heat treatment will not produce cracks in the metal and it can be welded or forged. Parts machined from this steel have satisfactory finish characteristics. Stocks are available in flats, rounds, squares, hexagons, octagons and billets of various sizes.



Midget Electric Motors

United States Instrument Corp., Electrotor Div., Summit, N. J.

The "gap-ring" armature used in Electrotor midget electric motors permits the production of assemblies of $\frac{3}{16}$ in. length and diameter weighing less than a gram. Type 240, illustrated, is the smallest of three larger models that have practical application to mechanical toys, motion picture cameras and projectors, air conditioning units, electric shavers and radio-controlled devices. Conventional commutator and segmentary winding are replaced with a "gap-ring" armature winding mounted on a bobbin and spindle that rotates within a permanent ring magnet. The armature can be considered a horse-shoe electromagnet in which broken iron rings form the core and the winding, the magnetic coil. A small dry cell battery energizes the coil. Type 240 operates at 5,000 rpm on 3 to $4\frac{1}{2}$ v, measures 9/16 in. in width and $\frac{7}{8}$ in. in diameter and weighs $\frac{3}{4}$ ounce.



Small Cooling Turbine

AiResearch Mfg. Co., 9851 Sepulveda Blvd., Los Angeles, Calif.

A small turbine refrigeration unit, weighing less than 5 lb, has been produced for cooling the interiors of jet

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Crane Co., 836 S. Michigan Ave., Chicago 5, Ill.

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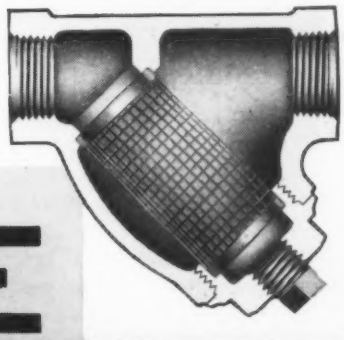

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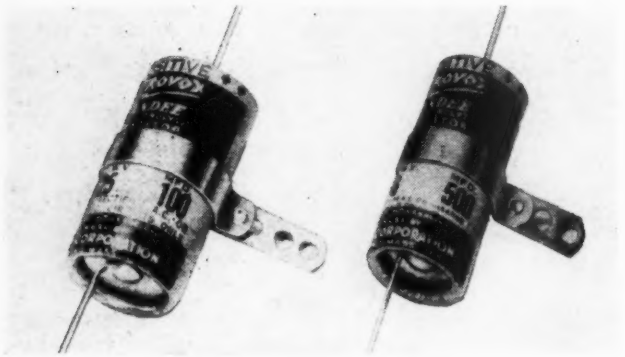
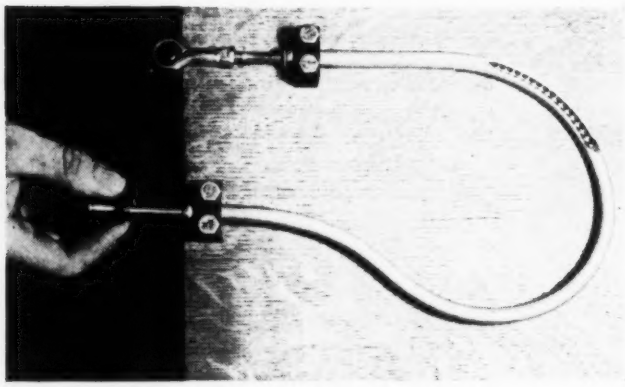
propelled planes. Using compressed air as the only source of power, the turbine can drop air temperature 135 F at 7 lb per min air flow. In combination with an 11½ lb heat exchanger, the unit will create a 500 F temperature drop. Only moving part is the rotor that includes fan and connecting shaft. Rotor revolves at speeds in excess of 100,000 rpm and will withstand a maximum speed of 140,000 rpm. Wind velocities within the unit have been observed up to 800 miles per hour.



Midget-Can Electrolytics

Aerovox Corp., New Bedford, Mass.

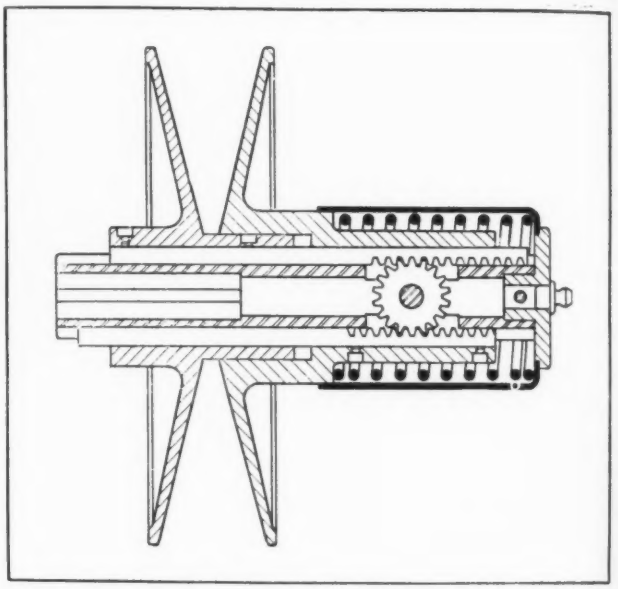
Twelve high-capacitance low-voltage units have been added to the Type PBS midget-can Dandee line of electrolytics. The new units are used in electric fence assemblies and in other applications requiring very high capacitance at low voltages. Standard Dandee construction is used throughout: hermetically-sealed metal can, generous-length tinned wire leads and center mounting strap. The low voltage ratings are—6, 12, 15, 18, 25, 50 v d-c working; high capacitance ratings are—100, 250, 500, 1,000, 1,500, 2,000, 3,000, 4,000 microfarads.



Push-Pull Controls

Southwest Products Co., 10 S. Catalina St., Pasadena 1, Calif.

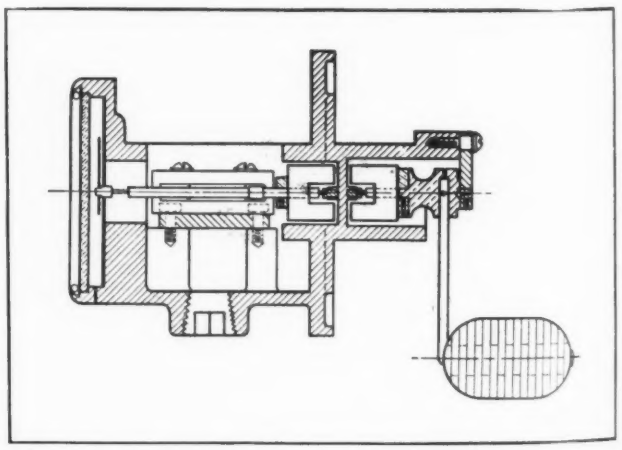
Absolute manual remote control of mechanical, hydraulic or other devices is provided by a line of push-pull controls. Three types are available: (1) light duty—⅜ in. OD rigid housing, load range to 1,250 lb ultimate compression; (2) heavy duty—7/16 in. OD rigid housing, load range to 1,650 lb ultimate compression and (3) extra heavy duty—½ in. OD rigid housing, load range to 3,050 lb ultimate compression. The moving element comprises a flexible tinned aircraft cable over which a series of steel or stainless steel spherical shells are strung, as shown in cut-away portion of the illustration. Standard "O" rings seal both ends of the aluminum or stainless steel housing. A bend radius of 2 in. allows for ease of installation under almost any condition.



Variable-Pitch Pulleys

Gerbing Mfg. Corp., 154 E. Erie St., Chicago 11, Ill.

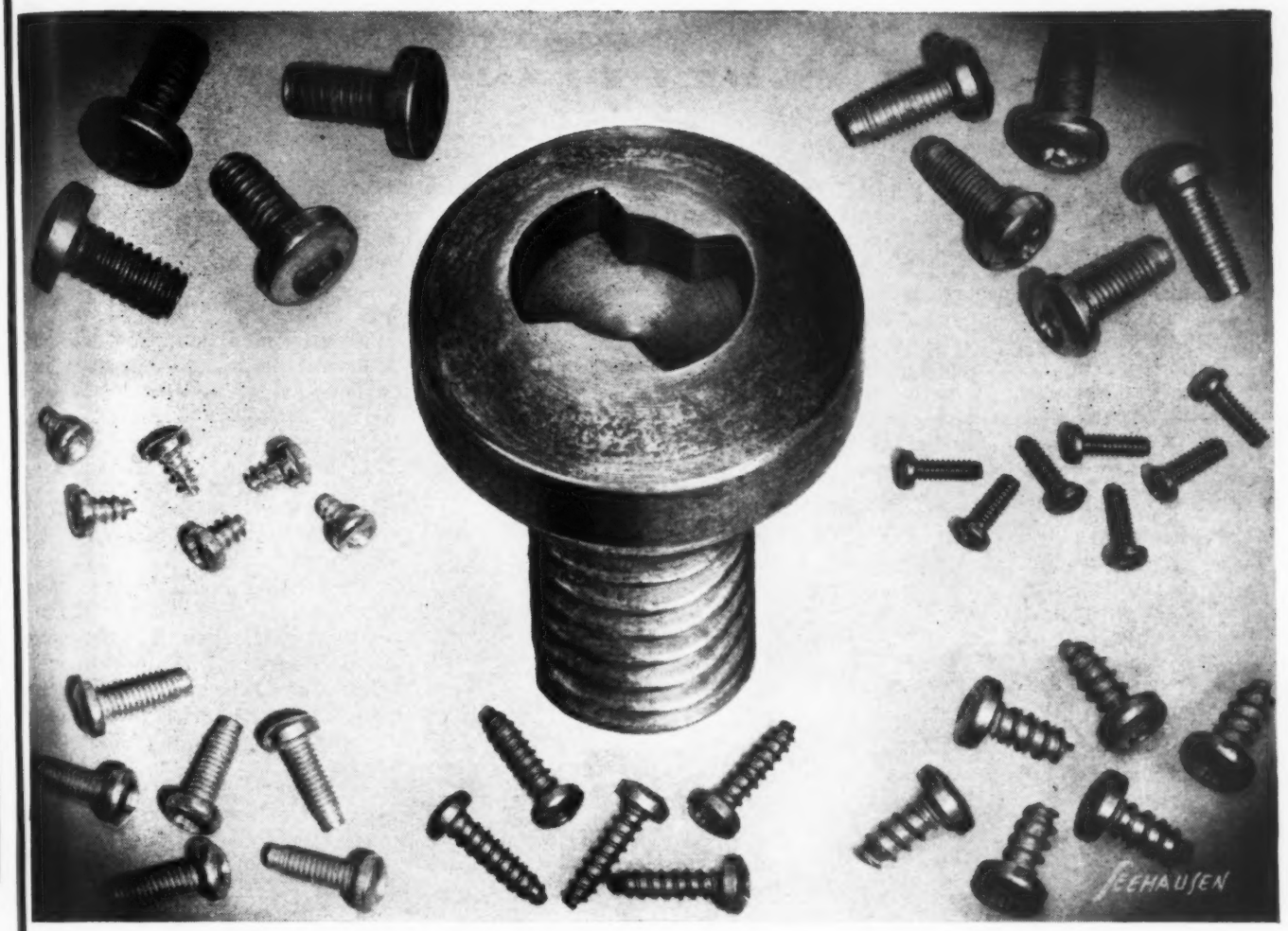
Roto-Cone variable-pitch pulleys are designed around a double rack and pinion mechanism that keeps movement of the two sheave faces equal and opposite at all times. A single spring provides the necessary pressure for proper belt tension. As a constant belt centerline is maintained, V-groove driven sheaves can be used. Electronic balancing of the complete assembly during manufacture reduces operational noise and wear. Horsepower ratings vary from ⅛ to 7½, speed variations within a 3 to 1 ratio.



Liquid Level Gage

Boston Auto Gage Co., Pittsfield, Mass.

Developed for indicating the level of insulating liquid in transformer tanks, this gage utilizes two General Electric sintered alnico 2 permanent magnets. An aluminum



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Since its adaptation to the Clutch Recess, the versatility of the Pan Head, in structure and finished appearance, has met with such popular acceptance by CLUTCH HEAD users that we recommend its investigation from the angles of economy and greater efficiency.

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diaphragm pressure tight to a minimum of 30 psi separates the magnets and provides an effective seal between the liquid and the gage proper. Motion of the liquid is transmitted from the float through the arm to one of the magnets. This magnet, in turn, transfers its motion by magnetic force to the second magnet that actuates a dial indicator needle. To effect a seal in the opening where the gage is installed, the flange is mounted on the tank with four studs, usually below the maximum liquid level, and sealed with a hy-car gasket.

Pneumatic Transmitter

The Foxboro Co., Foxboro, Mass.

The Model 42 pneumatic transmitter incorporates a completely redesigned transmitting element that is exactly duplicated in the receiving instrument. Precise matching of the two mechanisms combined with simplified linear calibration gives improved transmission accuracy. Industrial process measurements including flow, static or differential pressure, liquid level, temperature or humidity are some of the applications of the instrument. The transmitting element is available with either eccentric, as illustrated, or concentric scale. The case is 12 in. in diameter and provides room for additional units as alarm or signal devices.

Molded Shaft Seals

Brummer Mfg. Corp., 1324 S. McKinley St., Chicago Heights, Ill.

Two models of a molded synthetic rubber shaft seal have been developed. A pressure fit around the shaft is achieved through the use of a steel torque band in combination with a specially designed compression spring. Both are contained between a treated fabric liner and the rubber body. The cartridge type unit, used primarily as a water pump seal, has an outer steel housing. The larger plain split-ring type is used as a rear main bearing seal in several automobile engines.

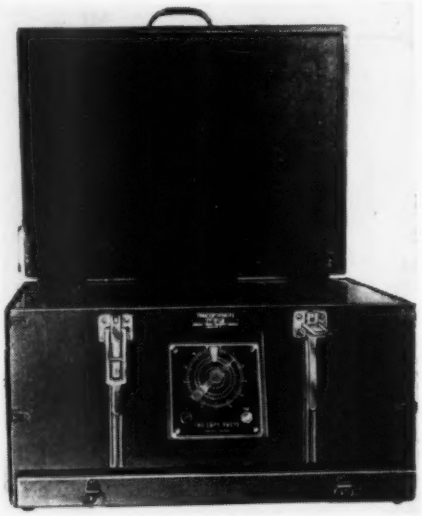

Photo-Copying Machine

General Photo Products Co., General Photo Building, 15 Summit Ave., Chatham, N. J.

Quantity reproduction of printed or written copy and photographs can be accomplished with the Tru-Copy-Photo copying machine. No focusing, heating or dark room is involved and uniform results are assured through the use of the automatically-set electric timer on the front of the case. The copying surface measures 14 x 20 in. and is made of Tufflex unbreakable glass. Unit is furnished with all necessary accessories including trays, paper, chemicals, color filter, red pilot light and squeege board.

Midget Instruments

International Instruments, Inc., New Haven, Conn.

The accuracy and dependability associated with standard size instruments is claimed for a line of midget meters that will fit a 1 in. dia opening. Of D'Arsonval design, the meters are obtainable in six basic ranges: 0-100, 0-200, 0-500 micro amp and 0-1.0, 0-5.0 and 0-10.0 ma. Shunted ammeters, a-c and d-c voltmeters are also available, self-contained in cases of slightly larger dimensions. These instruments find application in fields including electronic equipment and aircraft control panels.

Ultra High-Speed Motor

Bryant Chucking Grinder Co., Springfield, Vt.

A three-phase induction type motor that runs at a speed of 204,000 rpm on 3,400 cycle current is in the process of development. Designed for incorporation in the spindles of internal grinders, the motor has specific utility for grinding holes of 1/16 in. diameter or less. The rotor surface travels at approximately 600 mph and exerts an outward force measured in tons. Made integral with the grinding wheel spindle, the rotor is carried on precision ball bearings that are oil-mist lubricated. A water jacket is used for cooling.

Time Switches

Sangamo Electric Co., Springfield, Ill.

Simple adjustment of two levers sets the Type S time switch for automatic "on" and "off" operation. External manual control can also be affected without disturbing the automatic settings. The dial is calibrated in 15-minute intervals and accommodates a 24-hour period. Actuation is from a high torque, synchronous, self-starting motor that runs at 450 rpm on

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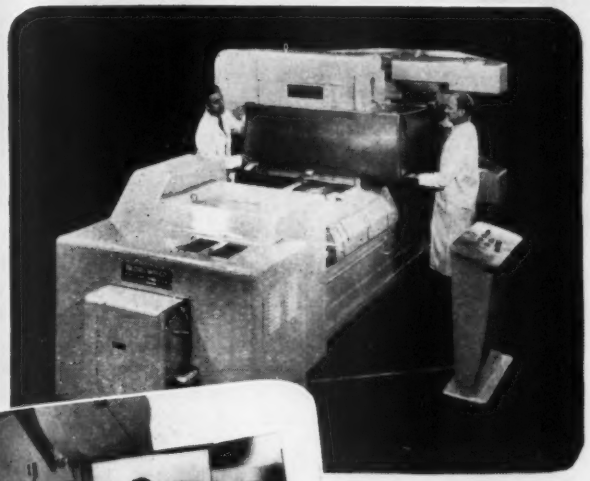

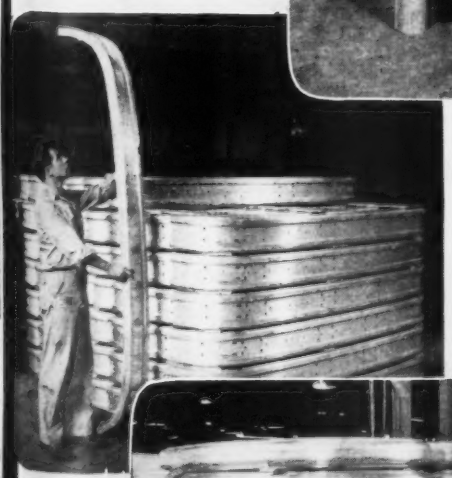

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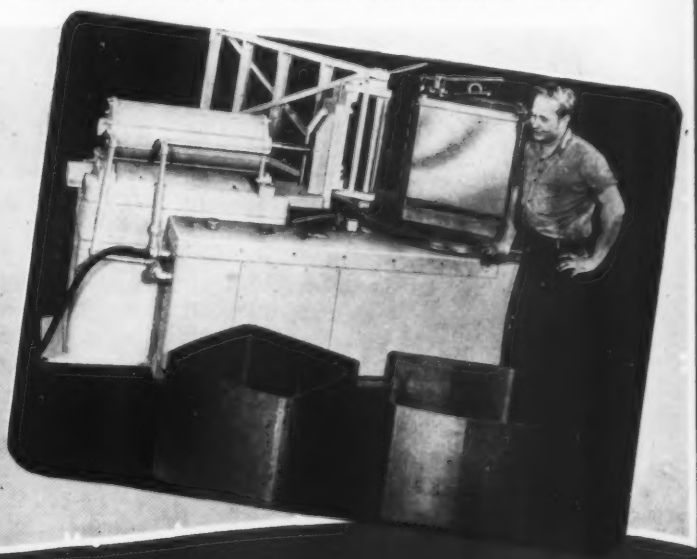
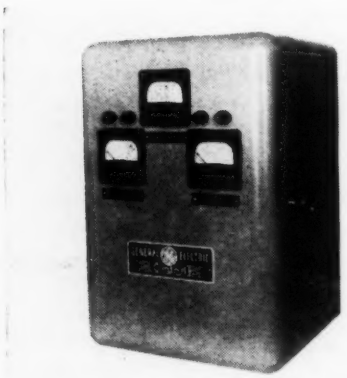


120 v, 60 cycle current. Rated at 15 amp, single pole, single-throw silver contacts are operated by a lever action that provides positive quick-make and quick-break. The type SR switch is similar and has two "on" and two "off" operations. Approximate dimensions are 3 x 5⅜ x 3 inches.

Ribbed Rubber Mat

Mid-West Development Co., 2185 Scott Lake Rd., R. D. 9, Pontiac, Mich.

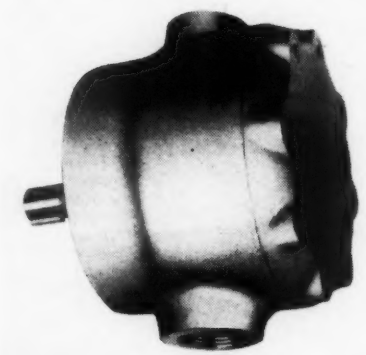
An improved flexible mat, Sani-Flex, is made of high grade neoprene for applications where a fire retardant material is required or of durable rubber for general applications. In addition to its use for standing mats to relieve fatigue of workers, it is ideal as entrance mats to buildings and homes, as standing mats in wet places and as a vibration damper under machinery. It also has been applied in the pits of bowling alleys to absorb ball and pin impact and against surfaces requiring padding of a flexible nature. Sheets are available in brown and black colors up to a maximum size of 31 x 46 inches.



Amplidyne Control Unit

General Electric Co., Schenectady 5, N. Y.

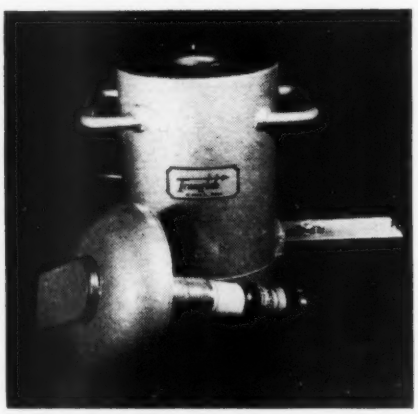
A packaged amplidyne control unit designed for use on steel mill and similar mill and processing line control applications where speed, current, or voltage regulation is required. The unit brings together in one 18 x 12 x 11 in. package all the adjusting rheostats, capacitors, rectifiers and instruments required for the amplidyne. In addition to performing all the necessary functions associated with regulating control circuits, the equipment provides motor speed limit control, field forcing during acceleration and deceleration.



Cartridge Hydraulic Pump

Sundstrand Machine Tool Co., Hydraulic Div., Rockford, Ill.

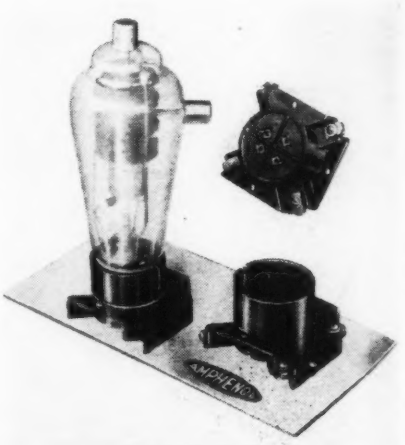
Built in three basic sizes, a constant displacement hydraulically balanced pump incorporates any one of four interchangeable cartridges that determine the capacity of each pump size. Twelve different capacities are thus available in a 2 to 75 gpm range. The cartridge sub-assembly simplifies pump inspection and service and provides means for quick change of pump rotation and capacity. As pump impulses are distributed to maintain complete pressure balance, bearing loading is reduced to a minimum. A system of ports connecting all pumping areas insures unrestricted flow at all times. Volumetric efficiency is said to be 90 percent with an operating pressure of 1,000 psi and speed of 1,200 revolutions per minute.



Shielded Sample Changer

Tracerlab, Inc., 55 Oliver St., Boston, Mass.

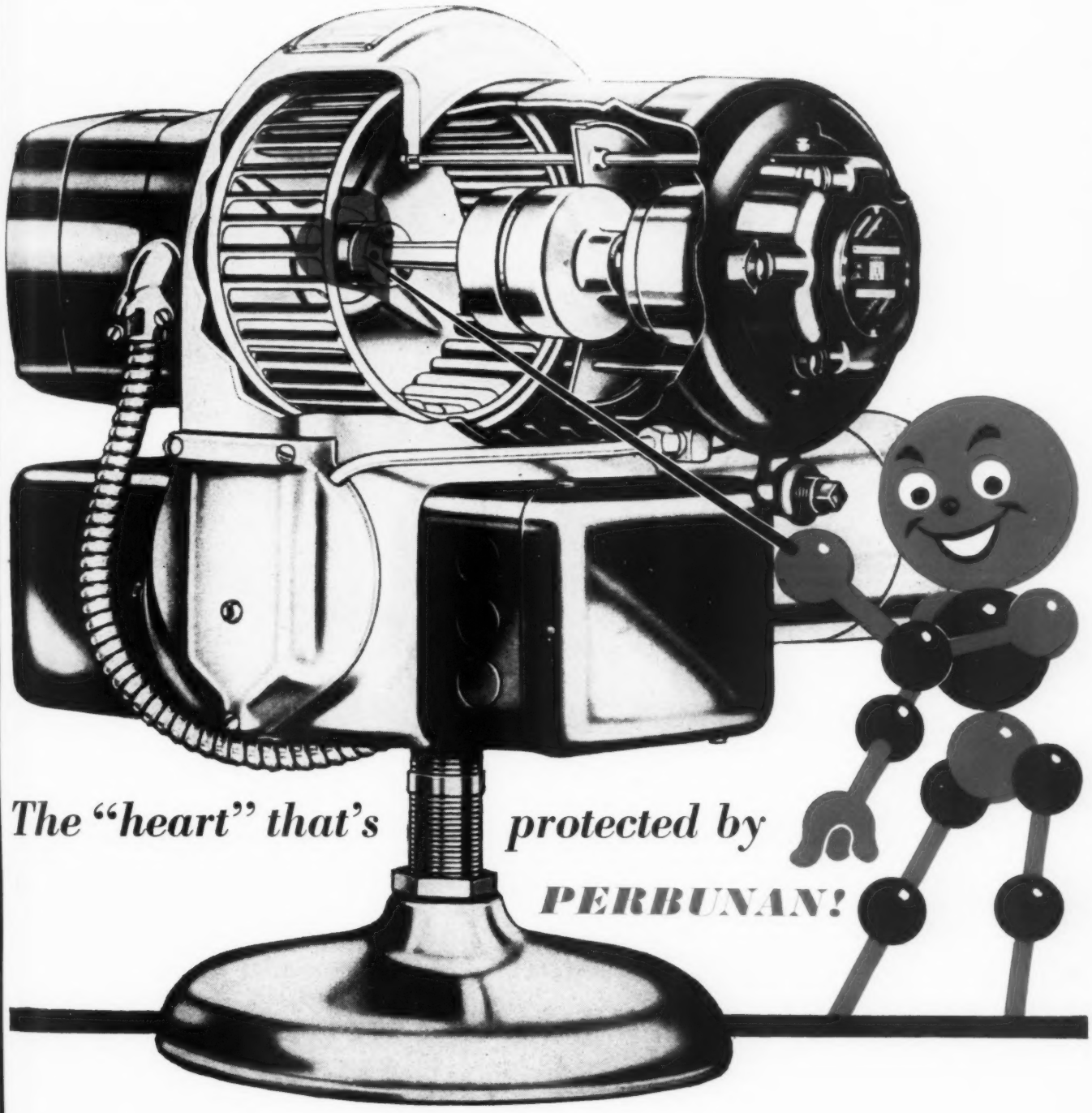
Laboratory personnel will be interested in the SC-9A lead-shielded manual sample changer and preamplifier that permits fast, accurate counting of a large number of radioactive samples. The unit comprises a two-inch lead shield 11⅝ in. high x 6½ in. diameter for lowering background count by a factor of three to five. The upper section of the shield holds a preamplifier circuit and Tracerlab G-M tube and a stainless steel sample holder is mounted on a slide in the lower portion of the shield. The sample holder is designed to take standard 1 in. diameter samples on planchets, filter paper and in ashing dishes, as well as standard radioactivity reference sources. Geometrical reproducibility within 0.1 percent is attained. Provision is also made for mounting calibrated foil absorbers between the sample and the G-M tube to determine energy level of the radiation being measured and to lower the detected count on samples whose activity is sufficiently high to introduce coincidence counting errors.



Industrial Tube Socket

American Phenolic Corp., Chicago

To meet the requirements of manufacturers of high quality electronic equipment, molded bakelite sockets have been developed. Design is based upon the electrical and structural requirements of all tube types requiring the RMA super jumbo tube base as well as industrial 411 and 412 tube bases that have the same pin arrangement but a smaller shell diameter. Socket contacts are of high conductivity material and of special cloverleaf design providing four lines of contact along the tube pin. Repeated tests indicate a contact resistance of considerably less than .001 ohm at current levels as high as 25 amperes. The units can be installed above or below horizontal mounting panels. Screw type terminals are used to facilitate wiring, maintenance and assembly and long surface creepage distances permit high voltage applications.



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• The heart of the revolutionary and patented new Gilbarco Oil Burner is the "Economy Clutch" that practically eliminates soot. It allows the burner's fan to spin freely before the oil pump is engaged to produce the flame, and again lets the fan spin after the burner has shut off.

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NEW BOOKS

Elementary Differential Equations

LYMAN M. KELLS, *Professor of Mathematics of the U. S. Naval Academy. Third Edition.* 312 *pages,* 5½ x 8 *in. Published by McGraw-Hill Book Company, Inc* 330 *West* 42*nd St., New York* 18, *N. Y.* $3.

Fundamental types of differential equations are presented with illustrative examples and numerous exercises. An illustrated geometrical discussion of solutions of differential equations is followed by a statement of simplified existence theorems. These theorems are later presented in more comprehensive form and illustrated with examples and comments to facilitate a detailed understanding.

In this third edition, a better balance has been attempted by introducing theoretical discussions and expanded treatments. Practically every article of the book has been revised to give maximum returns for time spent in study. Operators have been given more emphasis. Answers to exercises presented in the text are given in the back of the book.

Experimental Stress Analysis

PROCEEDINGS *of the Society for Experimental Stress Analysis, Vol. I, No. V.* 136 *pages,* 8½ x 11 *in. Published by Addison-Wesley Press, Inc., Kendall Sq., Cambridge* 42, *Mass.* $6. $6.

Collection of 15 papers. Titles of paper with respective authors are:

Explosive Impact Tests. Paul R. Shepler.

Criterion of Static and Fatigue Failures. Albert Sniderman.

Experimental Determination of the Dynamic Structural Response of an Airplane to Impact Loadings. W. M. Claflin.

A Proposed New Shock Measuring Instrument. W. P. Welch.

Radio Pulse Telemetering. Conrad H. Hoeppner.

Method of Obtaining the Stress at the Mid-Thickness by Measurements from Only One Surface of a Plate. A. Boodberg and E. D. Howe.

Bonded Wire Strain Gage Techniques for Polymethyl Methacrylate Plastics. A. G. H. Dietz and W. H. Campbell.

Discussion of Above Paper. W. J. Eney.

Effect of Temperature Gradients upon the Introduction of Residual Stresses in Weldments or Other Structures. J. R. Stitt.

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Method in the Field of Stress Analysis. J. T. Norton and D. Rosenthal.

Lateral Vibration and Stress in a Beam under Shock Machine Loading. E. G. Fischer.

High-Temperature Strain Gages and Their Application to Measurement of Vibratory Stresses in Turbosupercharger Buckets. S. S. Manson, R. H. Kemp, and W. C. Morgan.

Some Characteristics of Navy "High Impact" Type Shock Machines. Irwin Vigness.

Electronic Commutation of Strain Gages for Telemetering. Lawrence Lee Rauch.

Determination of the Effect of Ground Impact Forces in the Airplane Drop-Test. A. H. Pedersen and J. G. MacCarthy.

Magnetic Control of Industrial Motors

GERHART W. HEUMANN, *Control Engineering Div., Apparatus Dept., General Electric Company.* 589 *pages,* 6¼ x 9¼ *in. Published by John Wiley & Sons, Inc.,* 440 *Fourth Ave., New York* 16, *N. Y.* $7.50.

Performance of motors, characteristics of control devices, and the function of commonly used control circuits are explained. Considerations governing the selection of controllers to perform specific jobs are emphasized.

The 20 chapters include discussion of: Symbols and diagrams; d-c and a-c motors; d-c and a-c contactors and relays; accessories; basic control circuits; protection intermittent-duty controllers; and maintenance.

The Chemistry and Physics Of Organic Pigments

L. S. PRATT, 359 *pages,* 6 *x* 9¼ *in. Published by John Wiley & Sons, Inc.,* 440 *Fourth Ave., New York* 16. $6.

This book includes a rather broad background of history, and discusses theoretical considerations of the relation between color and chemical and physical structure. Covered are the raw materials for synthetic organic pigments and the chemistry involved in the formation of colored products, the typical chemistry of dyestuffs and organic pigments, systematic consideration of structural types, methods of testing known and unknown pigments, analytical procedures and identification and uses to which the organic pigments are adapted.

In a special chapter, emphasis is placed on a new interpretation of the significance of physical structure as related to color, a factor not ordinarily recognized and of little importance in the field of dyestuffs, but of funda-

mental importance in the field of organic pigment.

Applications for organic pigments discussed in the book include paint, varnish, lacquer, enamels, printing inks, carbon papers, typewriter ribbons, rubber, lineoleum, natural and artificial leathers, paper, wax, plastics, ceramics and cement.

1947 Guidebook and Directory for the Metal Finishing Industries

Annual Publication. 436 pages, 5¼ x 7⅞ in. Published by Metal Industry Publishing Co., Inc., 11 W 42nd St., New York 18, N. Y. $150.

This comprehensive reference manual on finishing processes combines the 16th edition of the Plating and Finishing Guidebook and the 6th edition of the Metal Finishing Buyers Directory. Subjects discussed in detail include: Abrasive methods, cleaning and pickling, electroplating solutions, surface treatments, control and testing, chemical tables, organic finishes and organic finishing methods. A directory of product classifications and a list of trade names are included. The book also includes an index and a list of supplier companies. Labelled thumb index separators are provided for quick reference to the various subjects.

The book will serve as a valuable reference for engineers, metallurgists and manufacturing personnel dealing with the application, specification and processing of metal finishes.

SPI Handbook

Society of the Plastics Industry, Inc. *451 pages, 6 x 9 in. Published by the Society of the Plastic Industry, Inc., 295 Madison Ave., New York 17, N. Y.* $4.50 *to members,* $7.50 *to non-members.*

This book gives engineers a source of standard information on plastics and the best practices of the plastics industry. The SPI work now assumes a place comparable to the handbook published by The Society of Automotive Engineers.

In preparation four years, three hundred technicians and other authorities from over 600 firms worked on the book. Six chapters were released as preliminary booklets in the last three years but these have been revised for the final publication.

Chapter headings of the book include: Classification of Plastics Molding Materials; Molding and Forming Plastics Parts; Design of Molded Articles; Design Standards for Inserts —Their Application in Plastics Parts; Standards for Tolerances on Molded Plastics Parts; Cementing and Assem-

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bly of Plastics; Testing Plastics Parts; Mold Design and Recommended Steels; Machining and Finishing Plastics Parts; Laminated Products and Their Fabrication.

Of special interest to design engineers is Chapter 3, "Design of Molded Articles". Six essential steps are outlined for the correct designing of a molded part. "Experience indicates that all too often when these precautions are not taken, (expensive) changes must be made after the molds are completed and the piece is in production."

The problems of shrinkage in plastics must be studied in preparing designs, and the amount a piece will shrink "is dependent upon one or a combination of the following factors: Chemical reaction, if any, in the plastic; temperature range over which it cools after its shape has been permanently established in mold, coupled with its coefficient of thermal expansion; and the degree to which material has been compressed in molding."

Tables and charts illustrate the effects of shrinkage in plastics, and precautions are outlined for avoidance of losses. Meeting the problem of distribution of stress in plastics, one section of the chapter deals with radii and fillets. An analysis of external and internal undercuts is outlined and illustrated.

Wall thicknesses of plastics are extremely important. Another section lists requirements of use—structure, weight, strength, insulation and dimensional stability; manufacturing requirements break down into molding—flow, setting, and ejection, and assembly—mainly strength and precision.

The importance of tapers or drafts is stressed, and the use of ribs "to increase rigidity and strength without increasing thickness." The reinforcement of holes in plastics articles through use of bosses, and the designer's problems in such apertures are covered in two sections.

The book makes an important point clear: "The designer must design specifically for molding. For example, it is impractical to duplicate by molding the appearance of a handle which has been fabricated from wood or metal by machining. To design a molded part of good appearance calls for knowledge of the molding technique and mold construction."

The "many different types of surface treatment to which plastics so readily lend themselves" are commended to the designer, and techniques of molded lettering and threading in plastics are covered in three additional sections.

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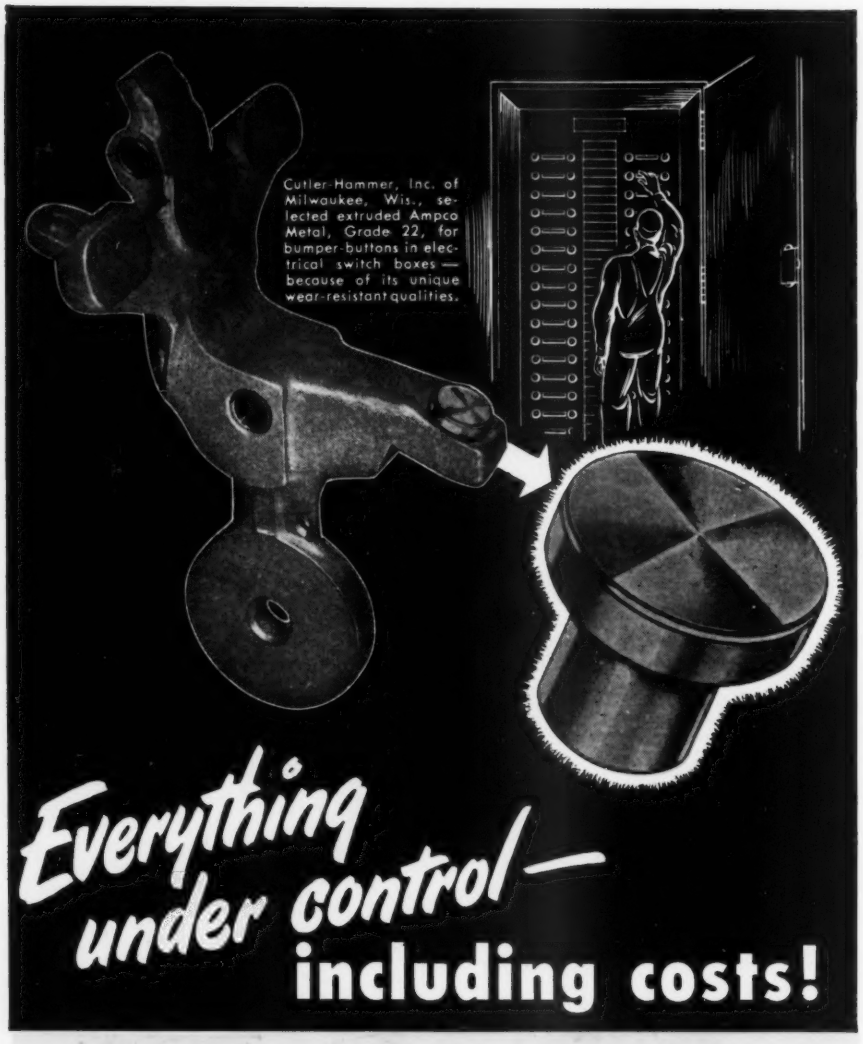
ENGINEERS AND BUILDERS OF OIL HYDRAULIC EQUIPMENT SINCE 1921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OUR READERS SAY

Drawing Is Important

To the Editor:

Industry has long stressed the importance of basic technical subjects such as engineering drawing in college curricula. The current trend in colleges, however, is toward a reduction in this basic training. In an effort to arrest this trend, Mr. F. G. Horton, General Director of the Engineering Society of Detroit, wrote Mr. F. A. Heacock, Chairman of the Drawing and Descriptive Geometry Division of the American Society for Engineering Education. Mr. Horton's letter contained the following resolution that was passed unanimously by the Board of Directors of the Detroit Society:

RESOLVED that The Engineering Society of Detroit advise the Engineering Council for Professional Development and the Drawing Division of The American Society for Engineering Education of its vigorous support of the retention of adequate training in Engineering Drawing in all Engineering curricula.

This is a very timely move and I hope the resolution will receive considerable publicity. — GEORGE E. ROWBOTHAM, *Project Engineer Fisher Body—Ternstedt Div., General Motors Corp., Member, Educational Committee E.S.D.*

Ed.—We second the motion heartily.

Industrial Design Courses

To the Editor:

With reference to your excellent article "How the Industrial Designer Works with the Engineer", we wish to express our deep interest in Industrial Design. This profession is practically nonexistent in this country. We have often felt the need of an expert to decide matters of eye appeal and the writer must rely on the views of salesmen who generally can only determine that the finished product "looks good" or not.

There is no course in any of our Technical Colleges or Universities on this subject, hence we would appreciate very much any guide you can give the writer of the best means to study this subject. Perhaps text books or a correspondence course with one of your colleges could be recom-

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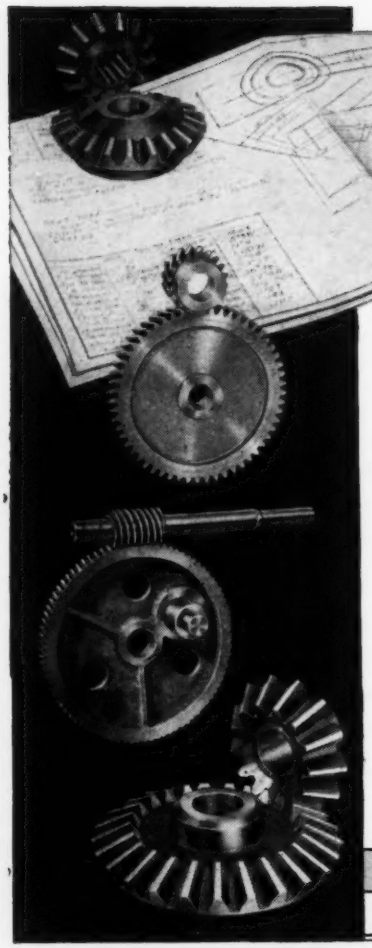

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mended. We would also appreciate an indication of the scope, time and fees of such a course at one of your universities, or recommendation to a suitable institute for us to pursue our inquiries.—C. C. MACNAUGHT, *Research Engineer, Moffat - Virtue Limited, Sydney, Australia.*

Ed.—Author Walter D. Teague has made the following comments in a letter to reader Macnaught:

"The profession of Industrial Design was just as rare in the United States 20 years ago as you say it now is in Australia, but it has established itself as an essential factor in American industry and no manufacturer today would think of bringing out an important product without expert design council.

"You may find useful my book "Design This Day; The Technique of Order in the Machine Age" which has just been published in a British edition by the Studio Publications, London, at a price of 35 shillings. It is a discussion of the basic principles of design and its text is illustrated with 128 plates. Another useful book is "Industrial Design, A Practical Guide," by Harold Van Doren, published by McGraw-Hill Book Company, Inc., N. Y. This deals with the actual processes of design applied to manufactured products.

"I am referring your letter to Philip McConnell, Executive Secretary of the Society of Industrial Designers, Inc., 48 E. 49th St., New York City, for information regarding the educational courses you mention. Many of our universities now have courses in Industrial Design, some of them leading to degrees. I do not know if any of them has a correspondence course but Mr. McConnell probably can inform you on this point."

Can You Help Them?

To the Editor:

Finland has an excellent and keenly scientific minded technical institute, Teknillinen Korkeakoulu. During the recent war its library was bombed and totally destroyed.

On a recent trip to Finland for the American Friends Service Committee, I discussed the situation with Dr. Martti Levon, Director of the Institute. He said he would welcome gifts of scientific and technical books and periodicals from America to take the place of those destroyed. In the remarkable efforts for recovery that the Finns are making, the lack of technical library facilities is a very serious handicap. It would be a practical act of freindship to a nation that holds America in high regard if Americans should contribute

What are your competitors doing about shortages?

A whale of a lot of them have converted to Kaiser Aluminum . . . and are *staying* converted.

But don't take our word for it. Read a few of their quotes, as reported by our men in the field.

1. STOVE MANUFACTURERS SAID:

"When we converted to sheet aluminum for our gas range griddles, we were afraid consumers would object to its lightness. Turned out they preferred it because it's a better heat conductor and has a fine appearance. That's why we're sticking with Kaiser Aluminum."

2. APPLIANCE MANUFACTURERS SAID:

"Nobody had to sell us on aluminum having plenty of customer appeal, but we were afraid of the cost. However, we found that on a unit cost basis, aluminum is just as cheap or cheaper than any other metal. Besides, we saved money on handling and shipping. We've converted to Kaiser Aluminum for keeps."

3. MANUFACTURERS OF HEATING AND VENTILATING EQUIPMENT SAID:

"When we switched to Kaiser Aluminum, our workers were enthusiastic. They liked its lightness, its cleanness, and above all, its workability. What's more, our customers found aluminum duct work far more efficient. We wouldn't think of changing back."

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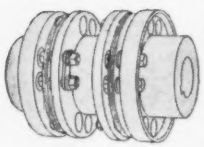

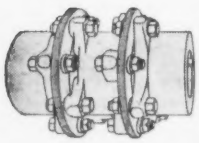

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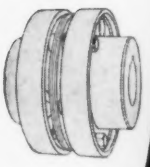

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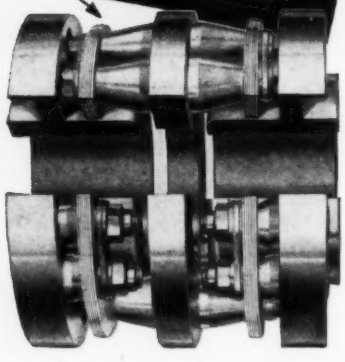

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—ARTHUR E. MORGAN, *Member American Friends Service Committee*

The Golden Mean

To the Editor:

Mr. Le Corbusier's "Modulor" system of measurement is of great interest to me.

The "Modulor" is essentially the system of the "golden mean" and the geometric connection with proportions of the human body is attributed to the German artist A. Durer. Mr. Le Corbusier represents his system in the same mystical form as did the ancient mathematicians but this is unnecessary. Taking your equation

$$A_2D_2 = D_2C_2 = 1$$

we have $\sqrt{5}$ as the diagonal length and

$$A_2E = \sqrt{5} - 1 = 1.236$$

making $EC_2 = 0.764$

thus $$\frac{A_2E}{EC_2} = \frac{1.236}{0.764} = 1.61$$

In modern usage, one might say that Mr. Le Corbusier wants to step sizes in the ratio of 1.61 and to introduce the series

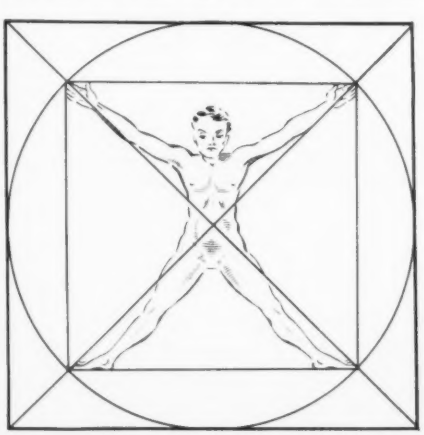
1 1.61 2.6 4.2 6.85 11.1

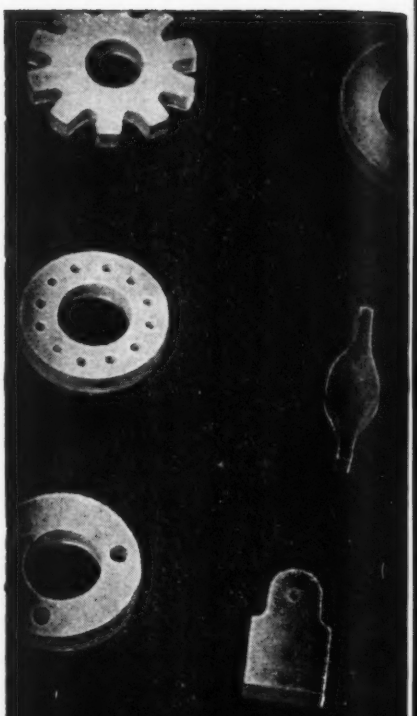
This series actually approximates the standard Preferred Number System of

1 1.6 2.5 4 6.3 10

which fits into the decimal system.

The connection of the human body with geometrical relations is quite natural for the builder and architect as he has to build for human beings and has to use dimensions fitted to them. This was already apparent to the ancients and the Roman architect Vitrunius described his system by placing a human being in a circle as illustrated. Obviously, the smaller square has half the area of the larger one. In modern





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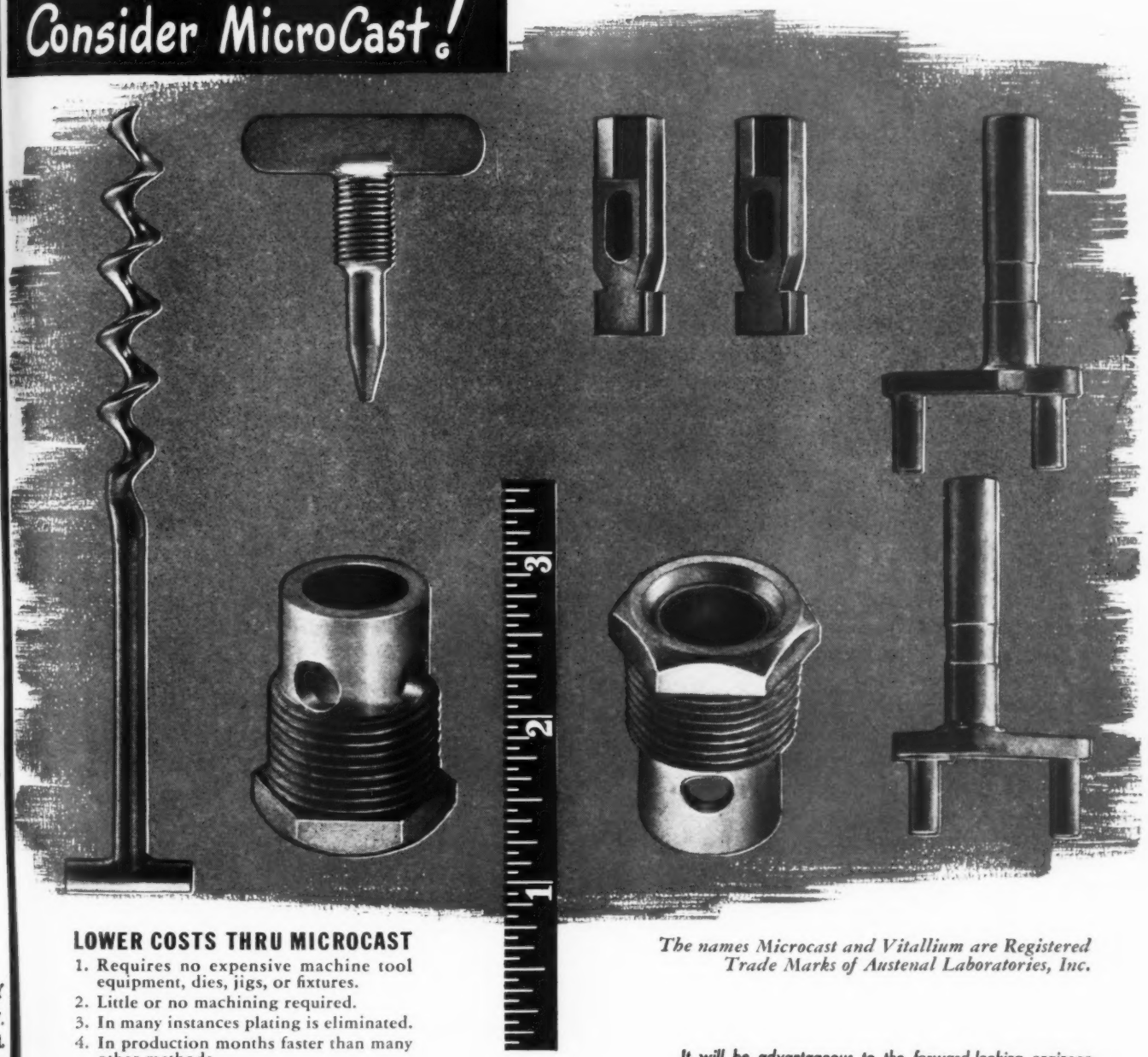

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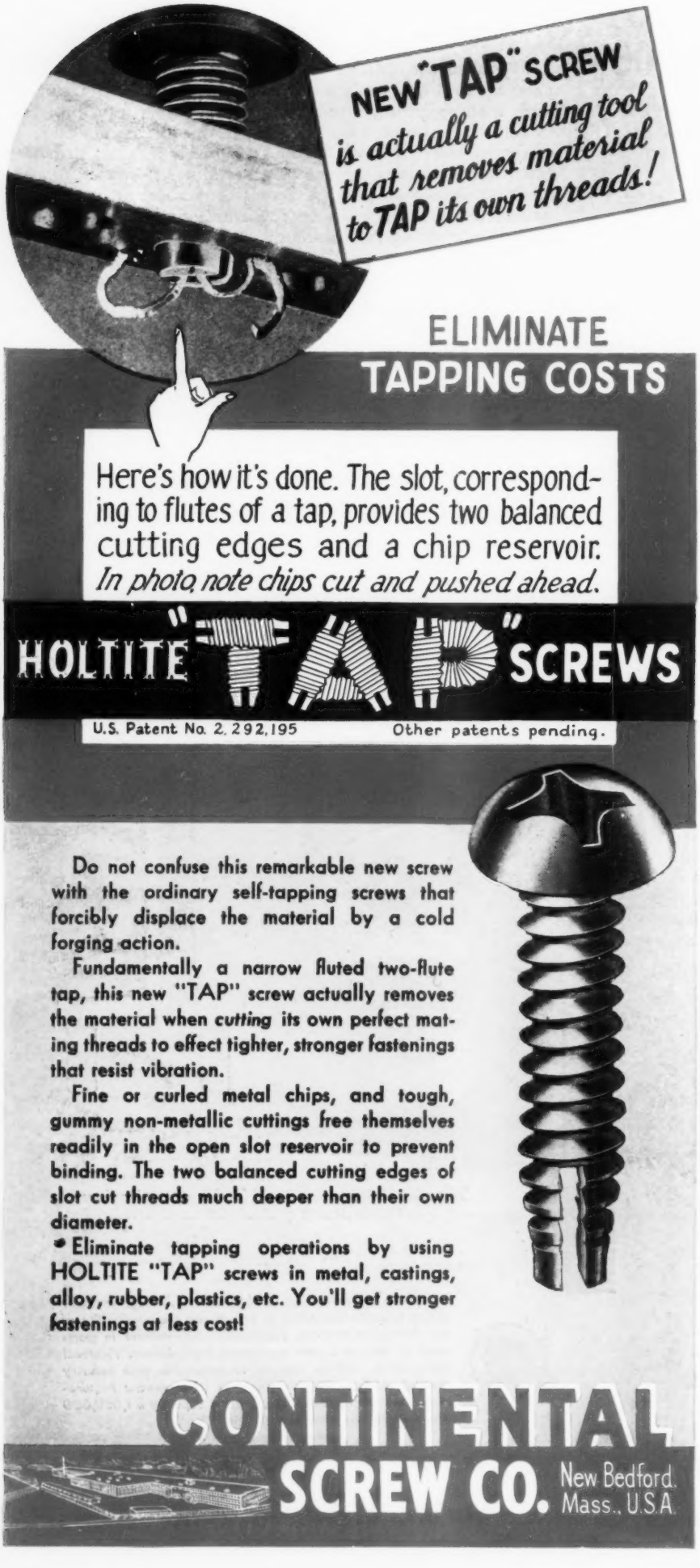
NEW "TAP" SCREW
is actually a cutting tool that removes material to TAP its own threads!
ELIMINATE
TAPPING COSTS
Here's how it's done. The slot, corresponding to flutes of a tap, provides two balanced cutting edges and a chip reservoir. In photo note chips cut and pushed ahead.
HOLTITE "TAP" SCREWS
U.S. Patent No. 2,292,195
Other patents pending.
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usage, we say that the length of one side is the $\sqrt{2}$ of the other. This leads to the following series.

1 $\sqrt{2}$ 2 $2\sqrt{2}$ 4 $4\sqrt{2}$ 8

which again approximates the Preferred Number System. This was well known to medieval architects, was used during the French Revolution for paper sizes and was re-invented some twenty years ago. Would not this system be preferable to the "golden mean"? Or, why not be still more liberal and modern? Instead of using one strict series, allow the Preferred Number System with its freedom of expansion and contraction to rule on designs.

—P. GRODZINSKI, *A.M.I. Mech. E. London, England*

Ed.—The standard Preferred Number System is slowly but surely finding more adoption. Why hesitate about using it?

Styled vs Designed

To the Editor:

I'm returning the release on the compressor—but puleeze, can't we say a little bit more about our designers? Something like this perhaps:

> Styling was done by the well-known industrial design team of Jinx & Blinks whose talents give to this product the maximum efficiency in a minimum of space.*

*If you think this is too long—you can always cut some of the technical stuff!—R. E. C.

To the Editor:

We still hate being credited as merely having "styled" the product. Can't you bring yourselves to use "designed"—since we are industrial designers, without any doubt?—D. R. S.

Ed.—But our engineering design readers insist that they do the designing. So do we.

The Alloys Are Better

To the Editor:

I have read with interest your Reference Book Sheets entitled "Application Tables to Guide in the Selection of Materials," in the November and December issues of PRODUCT ENGINEERING.

However, I believe a certain amount of confusion exists on page 165 of the December issue. Listed under cast iron applications is a group of specialized grades of cast iron: . . . "Meehanite, nickel, molybdenum and copper alloyed cast irons." Under

Redesign for Arc Welding Cuts Costs 42%

By W. Haddon Judson, President
Judson Brothers Co.
Collegeville, Pa.

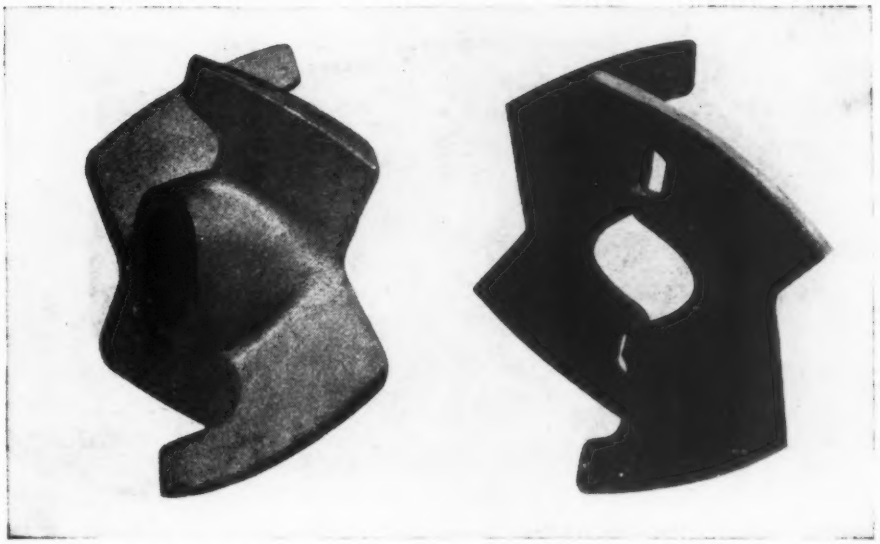

Fig. 1. Left: The former agitator blade unit. Right: The pressed steel unit, now used.

IN the manufacture of farm implements, we are searching constantly for production methods that will reduce costs and yet maintain high quality and wherever possible improve the strength and performance of our implements. One outstanding example of cost reduction is illustrated in the recent conversion to welded design for the agitator shafts of our lime and fertilizer spreaders.

A simple change in the design of one part for welded assembly has cut the shop costs of these agitator shafts from $6.96 to $4.03, representing a saving of 42% on these units alone. In addition to these cost savings, we have improved the performance of our product.

Originally, the agitator blade units were made from individual parts of conventional construction, each of which required several machining operations. The center hole had to be machined and a radial hole drilled and tapped. A set screw was then inserted for anchoring the blade unit to the shaft. With the welded design, the pressed alloy steel agitator blade units are slipped over the shaft and permanently assembled in place by tack welding them to the shaft. Hence, with this construction, the blade units become an integral part of the shaft, and this eliminates any possibility of the blades working loose in service. The welding is done with 3/16" "Fleetweld 37" electrode and Lincoln "Shield-Arc" welders.

The blade units are welded to the hub of the blades as shown in Fig. 2. This simply requires a tack weld between the bars and the hub of the blade unit. This design, made practical by welding, increases the thoroughness of the agitation, further improving the performance of our spreader. Attaching sweep bars with our former construction would have been a difficult and costly operation.

The same principles of welded construction used in fabrication of our agitator shaft assemblies, are used also in building the center bearing support plate by welding the center bearing to a sheared steel plate as shown in Fig. 2. The main framework and body of the spreader (Fig. 3) are also of arc welded steel construction.

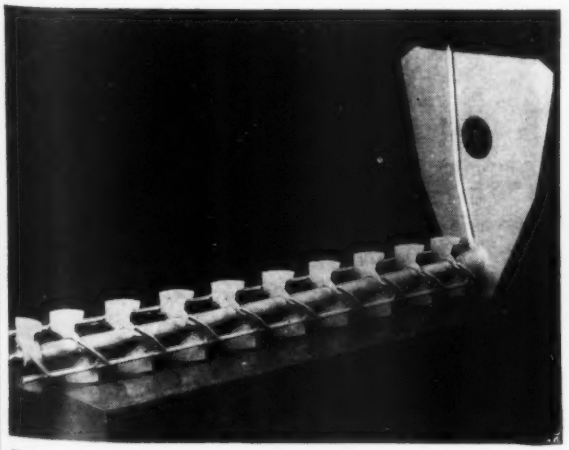

Fig. 2. Agitator shaft assembly showing welded blade units and sweep bars. All welded center plate is shown at right.

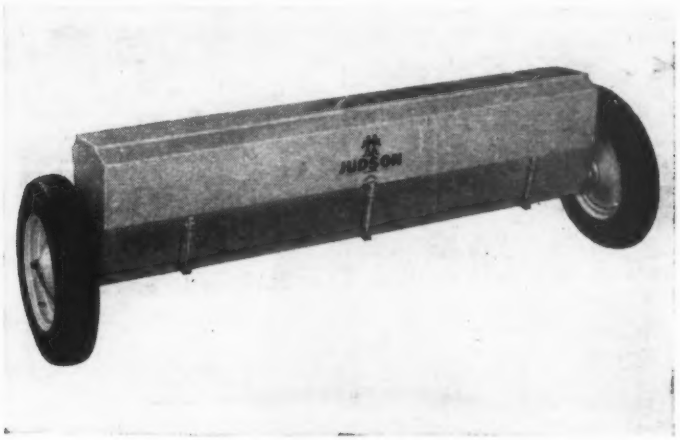

Fig. 3. All welded Judson lime and fertilizer spreader.

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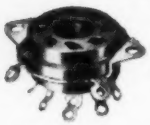
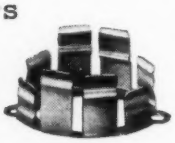


subsidiaries. They have helped many manufacturers



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remarks, is a list of qualities that do not apply to these specialized cast irons. Meehanite is hardly a low-strength material, and it is not brittle after heat-treatment. — RONALD A. WONDER, *Newark, N. J.*

Ed.—Reader Wonder is correct. The remarks pertain only to ordinary cast iron. The alloy cast irons including Meehanite possess markedly superior properties.

Many Thanks

To the Editor:

Let me congratulate you on good practical and spiritual sense in your editorial of October, 1947, "The Buncombe About Engineers and Politics." —FREDERICK FRANZ
New Haven, Conn.

To the Editor:

Allow me to thank you for your editorial "The Buncombe About Engineers and Politics". The engineering profession is suffering from educational time limitations caused by this buncombe.—CARROLL D. BILLMYER
Kingston, R. I.

Ed.—There are many more that feel about it just as keenly. More should go on record.

To the Editor:

In your November issue of PRODUCT ENGINEERING you have written an excellent editorial, entitled "Playing It Safe Is Often Dangerous."

Our staff frequently uses your magazine for reference. We feel it would be of great advantage to insert copies of this editorial in our student's note books.

—JOHN L. DICKINSON, *Chief Engineer*
Cal-Aero Technical Institute

Ed.—Glad you feel that way. By all means use the editorial. That's why we write them.

To the Editor:

Congratulations on your editorial "Some Are Merely Bricklayers." Your point is well taken and nicely stated. It should be required reading for every professional man. — W. R. HOUGH, *Chief Engineer, The Reliance Electric & Engineering Co.*

United We Stand

To the Editor:

The article "ECPD Raises Professional Standards of the Engineer" published in the Technical News Highlights section is most interesting.

I would like additional information

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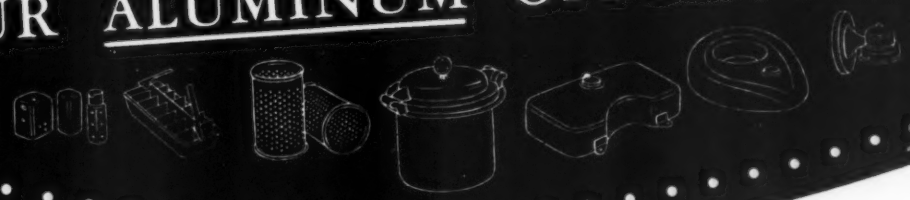

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regarding the Engineers' Council for Professional Development. Would you be kind enough to send me the full address?

May I take this opportunity to compliment you on the work of your excellent magazine. There are many engineering magazines, but PRODUCT ENGINEERING is the outstanding one, definitely in a class by itself.—CHESTER A. WOLFGRAM, *Milwaukee 4, Wis.*

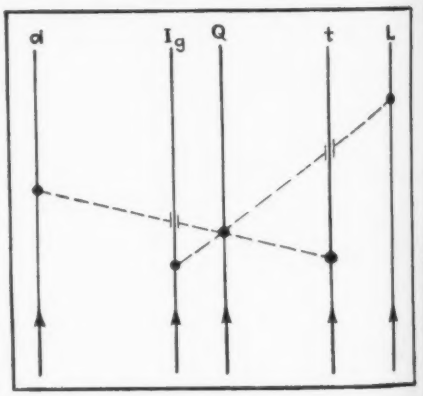
Ed.—The address of ECPD is: Engineering Societies Building, 29 W. 39th St., New York 18.

Diagram Clarified

To the Editor:

In nomograms of three parallel scales or combinations thereof, each computing secant cuts across all three scales. But when these are interlaced in combination, such as in the nomogram of H. F. Bariffi in PRODUCT ENGINEERING, July 1947, page 177, on Moment of Inertia by Torsional Pendulum, the significant intersections are not evident. To make them so, it would be advisable to use dots at these intersections and break the dashed lines of the secants at the other crossings, as in sketch. Then no text is needed for using the nomogram.

Unless there is some virtue in having the *L* scale run opposite to the



rest, it would seem best to have all scales increase in the same direction, preferably upward. Set up in this manner, there would be less likelihood of error in reading. The re-arrangement of scales in the sketch does this. Every nomogram should also show the algebraic equation on which it is based, and should include a sample calculation drawn upon it.

By suspending and timing the body in this way, together with the use of a solid comparison cylinder, the related radius of gyration *R* can be found; since $R^2 = \frac{I}{m}$ as in PRODUCT ENGINEERING, January 1946, page 88.

—CARL P. NACHOD
Louisville 4, Ky.